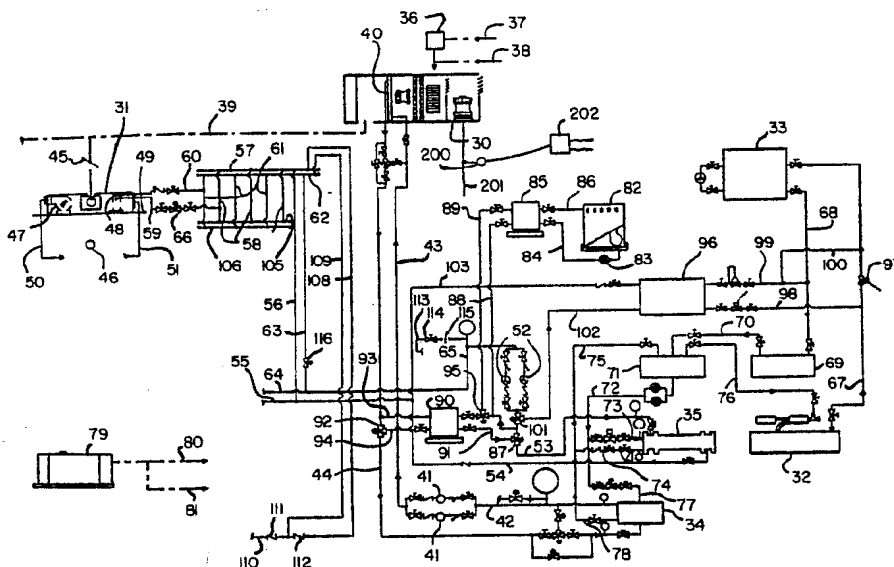




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(54) Title: AIR CONDITIONING APPARATUS**(57) Abstract**

The apparatus comprises a plurality of air outlets (50), means (30) for dehumidifying air, means (39, 45) for circulating dehumidified air to the air outlets, means (34, 40-44) operable to control its moisture content and temperature so that the dehumidified air is incapable, at the rate at which it is required for humidity controls, of maintaining the desired space temperature at the maximum design cooling load, a sensor (196, 197) for measuring the absolute or relative humidity of the space served by the outlets, means (195) responsive to the sensor, and operable to control the rate at which dehumidified air is delivered by each of the outlets to the space it serves, and means (45) operable to control the rate at which dehumidified air is delivered to the outlets.

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AIR CONDITIONING APPARATUS

BACKGROUND OF THE INVENTION

5

Field of the Invention

This invention relates to air conditioning apparatus and to a method for operating that apparatus. The apparatus is admirably suited for a building which has a sprinkler
10 system, an electrical grid, or both. Briefly, in one embodiment, the apparatus comprises an air handler, a plurality of induction-type mixing boxes, air circulating means, means for dehumidifying or for dehumidifying and cooling air circulated through the air handler, heat
15 transfer means for carrying a part of the air conditioning load, cooling means for transferring heat from a heat transfer fluid, a circulating system which preferably includes a part of the sprinkler system of the building for transferring heat from the heat transfer means to the
20 cooling means and at least one humidistat for controlling the apparatus. In a second embodiment the apparatus is a sprinkler system, and in a third comprises compression and absorption refrigeration apparatus, a cogenerator, a dehumidifier, a regenerator for the dehumidifier, a storage
25 tank, air circulating means and means for storing ice.

The Prior Art

Air conditioning apparatus for a building which has a sprinkler system, and which comprises an air handler, a plurality of induction-type mixing boxes, air circulating
30 means, means for dehumidifying air circulated through the air handler, heat transfer means for carrying a part of the air conditioning load, cooling means for transferring heat from a heat transfer fluid, and a circulating system which includes a part of the sprinkler system of the building for
35 transferring heat from the heat transfer means to the cooling means is suggested in "Westenhofer and Meckler", U.S. Patent No. 4,826,667, 1981 (see, also, "Meckler", U.S. Patent No. 4,033,740, 1977 and "Meckler (2)", U.S. Patent

No. 3,918,525, 1975). Such apparatus has been installed by The Social Security Administration in its Metro West Facility, Baltimore, Maryland, and in the Monroe County Court House, Stroudsburg, Pennsylvania (see Specifying Engineer, January, 1986).

A variable air volume induction mixing box which induces room air to temper, or plenum air to reheat, primary, conditioned air is suggested in "Meckler (3)", U.S. Patent No. 3,883,071, 1975.

10 The use of a cogenerator to produce both shaft work and heat has been suggested, for example by "McGrath", U.S. Patent No. 2,242,588, 1941; "Miller", U.S. Patent No. 2,284,914, 1942; "Meckler(4)", U.S. Patent No. 3,247,679, 1966; "Meckler (5)", U.S. Patent No. 3,401,530, 1968; and
15 "Meckler (6)", U.S. Patent No. 4,304,955, 1981.

Both Meckler (4) and Meckler (5) disclose apparatus which includes an internal combustion engine operatively connected to drive the compressor of compression refrigeration apparatus and means for conducting heat from
20 the engine to regenerate a chemical desiccant.

McGrath discloses a "heating system" which includes two compressors, both driven by an internal combustion engine for pumping heat in two stages from ambient air to a building. The internal combustion engine also drives an
25 electric generator and furnishes heat to the refrigerant of the heat pump. Heat is transferred to the refrigerant both from the exhaust gases of the internal combustion engine and from the cooling jacket thereof.

Miller discloses apparatus wherein the shaft of an
30 internal combustion engine drives both an electric generator and the compressor of compression refrigeration apparatus. The apparatus also includes means for transferring exhaust heat from the internal combustion engine to the desiccant of a regenerator of a chemical dehumidifier to provide heat
35 necessary for regeneration of the desiccant.

Meckler (6) discloses apparatus including an electric generator driven by an internal combustion engine and operation of the engine to supplement a solar collector, as

required, to provide heat for the regeneration of a chemical desiccant; the electricity generated when the engine is operated provides energy for pumps, blowers and the like of an air conditioning system.

5 Apparatus which heats a house by pumping heat from low temperature water and produces ice for subsequent cooling is disclosed by "Schutt", U.S. Patent No. 1,969,187, 1934.

Air conditioning apparatus in which a humidistat controls a humidified air valve is disclosed in British
10 patent No. 1,077,372. 1967, "Ozonair".

BRIEF DESCRIPTION OF THE INVENTION

This invention relates to air conditioning apparatus which is admirably suited for a building having a sprinkler system, an electrical grid, or both. In one embodiment, the
15 apparatus comprises a plurality of air outlets, means for dehumidifying air, means for circulating dehumidified air to the air outlets, means operable to control its moisture content and temperature so that the dehumidified air is incapable, at the rate at which it is required for humidity
20 control, of maintaining the desired space temperature at the maximum design cooling load, a sensor for measuring the absolute or relative humidity of the space served by the air outlets, means responsive to the sensor and operable to control the rate at which dehumidified air is delivered by
25 each of the air outlets to the space it serves to maintain the moisture content of the space served within control limits, and means operable to control the rate at which dehumidified air is delivered to the air outlets so that dehumidified air is available at the rate required from time
30 to time by each of the air outlets.

In another embodiment, the apparatus is an air outlet comprising an inlet for dehumidified air, means for delivering dehumidified air supplied to the inlet to a space to be conditioned, a sensor for measuring the absolute or
35 relative humidity of the space served by the air outlet, and means responsive to the sensor and operable to control the rate at which dehumidified air is delivered by the air outlet to the space it serves to maintain the moisture

content of the space served within control limits. In a preferred embodiment, the last-named means is also responsive to the pressure of the dehumidified air supplied to the inlet.

5 In still another embodiment the air conditioning apparatus comprises a mixing box to serve each of a plurality of spaces to be air conditioned, each of the mixing boxes having an outlet and a conditioned air inlet and being operable to deliver air through the outlet to
10 condition the space it serves and to cause air introduced into the conditioned air inlet to flow through the outlet to the space, each of the mixing boxes also having a heat transfer device and a fan operable to induce air to flow from the space, in heat exchange relationship with the heat
15 transfer device and then through the outlet to the space, the apparatus also comprising means for dehumidifying air, means operable to control its moisture content and temperature so that the dehumidified air is incapable, at the rate at which it is required for humidity control, of
20 maintaining the desired space temperature at the maximum design cooling load, means for delivering dehumidified air to the conditioned air inlet of each of the mixing boxes, means including a humidistat operable to increase or decrease the rate at which dehumidified air is delivered to
25 each of the mixing boxes as required to maintain a predetermined humidity, and means including a thermostat operable to increase or decrease the rate at which heat is transferred by each of the heat transfer devices to or from air circulated in heat exchange relation therewith to
30 maintain a predetermined temperature in each of the spaces.

BRIEF DESCRIPTION OF THE DRAWINGS

Fig. 1 is a schematic diagram of air conditioning apparatus according to the instant invention.

Fig. 2 is a fragmentary plan view showing a sprinkler
35 system that is an important part of the apparatus of Fig. 1.

Fig. 3 is a fragmentary plan view showing another embodiment of a sprinkler system that can be used in the apparatus of Fig. 1.

Fig. 4 is a schematic diagram of apparatus similar to that shown in Fig. 1, differing mainly in that certain components have been added.

Fig. 5 is a schematic diagram of apparatus similar to that shown in Figs. 1 and 4, differing mainly in that all components which are not absolutely essential, with the exception of a generator powered by a gas turbine, have been omitted.

Fig. 6 is a fragmentary plan view showing a sprinkler system that can be used in the apparatus of Figs. 1, 4, 5, 12, 14, 15, 16, 17, 19-26 and 33-35, and to produce a cooled plenum.

Fig. 7 is a fragmentary plan view showing a sprinkler system similar to that of Fig. 6, and additionally including a simplified mixing box and a cooled light.

Fig. 8 is a vertical sectional view of the simplified mixing box of Fig. 7.

Fig. 9 is a view in vertical section of the cooled light of Fig. 7.

Fig. 10 is a partially schematic view of a mixing box which is an important part of the apparatus of Figs. 1, 4 and 5, and additionally including an improved control.

Fig. 11 is a partially schematic view of a mixing box similar to that of Fig. 10 and additionally including a heat exchanger.

Fig. 12 is a schematic diagram of apparatus similar to that of Fig. 5 which additionally includes a mixing box that increases the energy efficiency of the apparatus.

Fig. 13 is a fragmentary plan view showing another embodiment of a sprinkler system that can be used in the apparatus of Figs. 1, 4, 5, 12, 14, 15, 16, 17, 19-26 and 33-35.

Fig. 14 is a fragmentary diagram of the apparatus of Fig. 4 showing connections for heating incoming air.

Fig. 15 is a schematic diagram of apparatus that is similar in function to that of Figs. 4 and 14, except that heat from a cogenerator is used to regenerate a liquid desiccant, and the desiccant is used in two stages of

chemical dehumidification to dehumidify air that is circulated to the space to be conditioned.

Fig. 16 is a schematic diagram of apparatus that is similar to that of Fig. 15, except that the first stage of 5 chemical dehumidification uses a solid desiccant.

Fig. 17 is a schematic diagram of apparatus in which a heat engine drives an electric generator and furnishes heat both to regenerate a chemical desiccant and as an energy source for absorption refrigeration apparatus.

10 Fig. 18 is a partially schematic view of an air handler which is particularly advantageous as an element of the apparatus of Figs. 1, 4, 5, 12, 14, 19-23 and 33-35.

Fig. 19 is a schematic diagram showing apparatus similar to that of Fig. 1, differing mainly in that it 15 includes mixing boxes of two different kinds.

Fig. 20 is a schematic diagram showing apparatus similar to that of Fig. 19, including mixing boxes of two different kinds, but differing from the Fig. 19 apparatus because the mixing boxes are different.

20 Fig. 21 is a schematic diagram showing apparatus similar to that of Figs. 5 and 12, but differing in that compression refrigeration apparatus has been omitted and an absorption chiller/ heater has been added.

Fig. 22 is a schematic diagram showing apparatus 25 similar to that of Figs. 5 and 12, but differing in that compression refrigeration apparatus has been omitted and a closed circuit evaporative condenser has been added.

Fig. 23 is a schematic diagram showing apparatus similar to that of Fig. 1, differing mainly in that a 30 different mixing box has been substituted for that shown in Fig. 1.

Fig. 24 is a schematic diagram showing apparatus similar to that of Fig. 21, but differing in that a combination comprising compression refrigeration apparatus 35 driven by a gas engine and a chemical dehumidifier has been substituted for compression refrigeration apparatus.

Fig. 25 is a schematic diagram showing apparatus similar to that of Fig. 4, but differing in that a reheat

coil has been added to a mixing box which is an essential part of the apparatus.

Fig. 26 is a schematic diagram showing apparatus similar to that of Fig. 25, but differing in that a cooling coil has been omitted from the mixing box which is an essential part of the apparatus.

Fig. 27 is a schematic diagram showing apparatus similar to that of Fig. 25, but differing in that reheat and cooling coils of the mixing box have been replaced by heat pipes.

Fig. 28 is a schematic diagram showing apparatus similar to that of Fig. 27, but differing in that one of the heat pipes has been omitted from the mixing box.

Fig. 29 is a schematic diagram in elevation showing details of one of the mixing boxes of the apparatus of Fig. 19.

Fig. 30 is a schematic diagram in plan showing further details of the mixing box of Fig. 29.

Fig. 31 is a schematic diagram in elevation showing another embodiment of a mixing box similar to that of Fig. 29.

Fig. 32 is a schematic diagram in plan showing further details of the mixing box of Fig. 31.

Fig. 33 is a schematic diagram showing apparatus similar to that of Fig. , but differing in that it shows another embodiment of a mixing box.

Fig. 34 is a schematic diagram showing apparatus similar to that of Fig. 20, but differing in that cooling and reheat coils have been added to mixing boxes of the dual duct type which are a part of the apparatus.

Fig. 35 is a schematic diagram showing apparatus similar to that of Fig. 25, differing in that a humidistat which measures the moisture content of return air has been omitted and thermostat/humidistat-controllers associated with each of several mixing boxes have been added.

DESCRIPTION OF PREFERRED EMBODIMENTS

Preferred apparatus according to the invention shown in Fig. 1 comprises an air handler 30, a plurality of mixing

boxes 31 (one of which is shown in Fig. 1) and refrigeration apparatus which includes a compressor 32, an evaporative condenser 33, and two different evaporators, one which serves an ice storage tank 34 and one which serves a water 5 chiller 35. The evaporator which serves the ice storage tank 34 operates to produce ice when its operation does not increase the demand charge, for example, on night cycle when the building served by the apparatus is unoccupied, while the evaporator which serves the water chiller 35 operates 10 when it is needed, for example, on day cycle.

Outside air can be directed through or by-passed around an indirect evaporative cooler 36, as indicated by arrows 37 and 38, before it is conditioned in the air handler 30 and distributed through risers (not illustrated) and ducts (one 15 of which is shown in Fig. 1, designated 39) to the building. In the air handler 30, air is conditioned by contact with a coil 40 to a dry bulb temperature of substantially 42°F. (6°C.). Ice water from the ice storage tank 34 at, say, 38°F. (3°C.) is circulated by pumps 41, flowing through a 20 line 42, the pumps 41, a line 43, the coil 40 and a line 44 back to the tank 34. The flow of ice water through the coil 40 is modulated to maintain the 42°F. (6°C.) temperature of the conditioned air leaving the air handler 30. Whenever the ambient air has a low moisture content, it is economically 25 desirable to use the indirect evaporative cooler 36 and, thereby, to reduce the requirement for ice water in the coil 40.

Conditioned air from the ducts 39 is delivered to the mixing boxes 31 at a rate which varies, depending upon the 30 settings of individual dampers 45, each of which is actuated by a thermostat controller 46. The mixing boxes 31 are of the "fan/coil" type, having constant speed fans 47 and coils 48. The fans 47 have a capacity greater than the maximum flow of conditioned air to the mixing boxes 31 when the 35 dampers 45 are in their full open position; as a consequence, air is caused to flow from a space served thereby into each of the mixing boxes, mixing with conditioned air, and being returned to the space from the

fan discharge mixed with conditioned air. The spaces served by the mixing boxes 31 are below, while the boxes 31 are above, ceilings 49. The air flow described above is indicated in Fig. 1 by an arrow having a head 50 which represents the flow of a mixture of conditioned air and recirculated air from the mixing box 31 and a tail 51 which represents the flow of air from the space into the mixing box 31.

Chilled water flows through the coils 48, being circulated by pumps 52 through a line 53, the water chiller 35, a line 54, a main header 55, a supply line 56, a header 57 of a first sprinkler grid, one of several sprinkler conduits 58 of the first sprinkler grid, a supply line 59, the coil 48, a return line 60, one of several sprinkler conduits 61 of a second sprinkler grid, a header 62 of the second sprinkler grid, a return line 63, a main return 64 and a line 65 back to the pumps 52. The chilled water circulated through the coils 48 is at a comparatively high temperature, sufficiently high that moisture is not condensed when room air at design conditions flows over the coils 48. In a typical instance, the water in the coils 48 will be at 58°F. (14°C.), and the room air will be at 75°F. (24°C.) and 50% relative humidity. In one embodiment of the instant invention, the capacity of each of the fans 47 is such that, when the air conditioning load is at the maximum design load, the associated damper 45 is in its full open position, and chilled water is flowing through the associated coil 48 at its maximum rate (as subsequently discussed in more detail), from 40 to 60% of the air conditioning load is carried by conditioned air and the rest of the load is carried by the coil 48. It has been found that, when the apparatus has these design parameters, significant savings are possible because the sizes of the ducts and blowers required to circulate conditioned air can be minimized. In a typical installation, the savings which can be realized by minimizing duct and blower sizes are nearly sufficient to offset the extra cost of the mixing boxes 31 and of the refrigeration apparatus including the

compressor 32 which has the capability of making and storing ice and of providing chilled water when needed.

As is subsequently described in more detail, the thermostat controllers 46 control the operation of the mixing boxes 31 by modulating the dampers 45 and valves 66 as required to maintain the humidity and temperature desired in the spaces served thereby. Ordinarily, it is necessary to maintain some minimum flow of ventilation air into the space being conditioned; as a consequence, the minimum setting for each of the dampers 45 is that setting which provides the minimum ventilation air, usually 0.10 to 0.15 cubic foot per minute per square foot of space served by a given mixing box. Accordingly, the system is designed for a minimum air conditioning load which can be accommodated by air from the duct 39 being delivered to the space at the minimum rate required for ventilation unless some expedient that is not illustrated in Fig. 1 is used to add heat to the air delivered by at least some of the mixing boxes 31. Heat can be added, for example, by unitary heat pumps, as subsequently discussed in more detail, or by circulating warm water through a second circulating system (not illustrated) to all or some of the coils 48 in the apparatus of Fig. 1.

As has been stated above, the refrigeration apparatus includes the compressor 32, the evaporative condenser 33, and two different evaporators, one which serves the ice storage tank 34 and one which serves the water chiller 35. On day cycle, the ice storage tank 34 contains a supply of ice sufficient to provide all the chilled water required by the coils 40 until the evaporator which serves the ice storage tank 34 is again operated. Only the evaporator which serves the water chiller 35 is operated, the refrigerant flow being from the compressor 32 through a line 67, the evaporative condenser 33, a line 68, a high pressure receiver 69, a line 70, a low pressure receiver 71, a line 72, a line 73, the water chiller 35, lines 74 and 75, the low pressure receiver 71 and a line 76 to the suction side of the compressor 32. The evaporator which serves the water

chiller 35 is controlled to maintain the required chilled water temperature in the coils 48 of the mixing boxes 31.

The refrigeration apparatus is also operated when the water chiller 35 is idle, but to produce ice. The refrigerant flow is from the compressor 32 through the line 67, the evaporative condenser 33, the line 68, the high pressure receiver 69, the line 70, the low pressure receiver 71, the line 72, a line 77, the ice storage tank 34, a line 78, the line 75, the low pressure receiver 71 and the line 76 to the suction side of the compressor 32. Enough ice is produced while the water chiller 35 is idle to provide all the chilled water required by the coils 4 during the next period of operation of the water chiller 35.

The apparatus of Fig. 1 is highly advantageous from the standpoint of the cost of energy (electricity) required for operation. It was designed to service an addition to a shopping mall which had an air conditioning system in which a mixture of ambient air and return air was cooled to a dry bulb temperature of 55°F. (13°C.), and the cooled air was circulated as required for sensible heat control. It is by comparison with the existing system that, as stated above, the savings which can be realized by minimizing duct and blower sizes are sufficient to offset the extra cost of the mixing boxes 31 and a substantial portion of the cost of the refrigeration apparatus including the compressor 32 which has the capability of making and storing ice and of providing chilled water. In the existing mall, the energy costs are divided about equally between the requirements for lighting and the requirements of the HVAC system. A "demand" charge, which is a flat monthly fee based upon the maximum rate of energy usage, is a substantial part of the energy costs for the HVAC system; the demand charge, of course, reflects the high cost of new generating equipment, which makes it highly desirable for a utility, for the country, to keep the maximum rate at which electricity is used as low as possible. The apparatus of Fig. 1 makes ice when there is no demand charge (because the usage of energy by the shopping mall, by the community served by the utility, is low), and

then uses that ice during the day to carry about one-half of the peak air conditioning load. While the refrigeration apparatus operates during the time when the electricity it uses contributes to the demand charge, its energy requirements during this time are less than half of the total requirements of the HVAC system. Furthermore, the apparatus includes a gas engine-generator 79 which can be operated to generate electricity to be supplied to the electrical grid of the building (not illustrated) as indicated by an arrow 80, to provide emergency power as indicated by an arrow 81, or both. It has been estimated that one-half of the cost of energy required by the HVAC system can be saved by using the Fig. 1 apparatus instead of duplicating the existing equipment. It is highly advantageous to operate the gas engine-generator 79 whenever such operation prevents an increase in "demand".

The apparatus of Fig. 1 also includes a cooling tower 82 and a pump 83 for circulating tower water from the cooling tower 82 through a line 84, through a plate and frame heat exchanger 85, and through a line 86 back to the cooling tower 82. Whenever the ambient humidity is sufficiently low to make it worth while, the tower 82 can be operated, and cooled water can be circulated therefrom to the heat exchanger 85 as just described for heat transfer with heat transfer fluid discharged from the pumps 52 and diverted by a three-way valve 87 to flow through a line 88, the heat exchanger 85, a line 89, a plate and frame heat exchanger 90 and a line 91 before entering the line 53 for flow to the water chiller 35 and to the coils 48 as previously described. If the water from the tower is sufficiently cold, it is not necessary to operate the water chiller; if not, reduced operation is sufficient. The apparatus also includes a three-way valve 92 which can be used to divert heat transfer fluid in the line 44 (returning to the ice storage tank 34 from the coils 4) for flow through a line 93, through the heat exchanger 90 and through a line 94 back to the line 44 for return to the ice storage tank 34. When heat transfer fluid is diverted to flow

through the heat exchanger 90, as just described, the valve 87 and a valve 95 can be used to divert the flow of heat transfer fluid discharged by the pumps 52 directly into the heat exchanger 90 for heat transfer to the fluid diverted 5 from the line 44 and flow through the lines 91 and 53 to the water chiller 35. Such operation may be advantageous whenever the ice in the ice storage tank 34 has excess heat absorbing capacity, beyond that required by the coil 40 to provide air at 42°F. (6°C.) for the rest of the day of 10 operation. Heat exchange between the two fluids, as described, reduces the requirement for refrigeration to provide water at 58° F. (14°C.) to serve the coils 48, and may eliminate that requirement altogether if the ice has sufficient excess capacity.

15 The apparatus of Fig. 1 also includes a heat recovery unit 96 which can be used on night cycle to provide warm heat transfer fluid, as required, for circulation to the coils 48. This is done by closing a valve 97 at least partially so that warm refrigerant from the compressor 32 20 flows from the line 67 through a line 98 to the unit 96, leaving the unit 96 through a line 99 and either flowing through a line 100 back into the line 675 or flowing directly into the line 68. In either event, there is warm refrigerant in the unit 96 from which heat can be 25 transferred to the fluid circulated by the pumps 52. This is done by setting a valve 101 to divert heat transfer fluid discharged by the pumps 52 for flow through a line 102 to the unit 96. After heat has been transferred thereto from the refrigerant in the unit 96, the fluid flows through a 30 line 103 to the main header 55 and then through whichever ones of the coils 48 serve spaces which require heat and back to the pumps 52 as previously described.

Further details of the sprinkler system of the apparatus of Fig. 1 are shown in Fig. 2 where the system is 35 indicated generally at 104. The header 57, an opposed header 105 and the sprinkler conduits 58 make up a first sprinkler grid, while the header 62, an opposed header 106 and the sprinkler conduits 61 make up a second sprinkler grid. As

previously described, the line 56 is connected to discharge heat transfer fluid into the header 57, while the line 63 is connected to receive heat transfer fluid from the header 62. There are sprinkler heads 107 spaced a given distance, which may be 10 feet (3.05 meters), from one another in the sprinkler conduits 58 and in the sprinkler conduits 61. The sprinkler conduits 58 are spaced a given distance, which may be 20 feet (6.1 meters), from one another, and the sprinkler conduits 61 are spaced the same distance from one another. Each of the sprinkler conduits 58 is spaced $\frac{1}{2}$ the given distance from one of the sprinkler conduits 61 or from two of the conduits 61, depending upon its position in the grid. Accordingly, the two grids jointly constitute a sprinkler system in which conduits are spaced from one another by, say, 10 feet (3.05 meters) and in which sprinkler heads in a given conduit are spaced from one another by, say, 10 feet (3.05 meters). Each grid, however, as is subsequently explained in more detail, is independently connected to a source for water to be used in case of fire, the first grid by a line 108 which is operably connected to introduce water into the header 57, and the second grid by a line 109 which is operably connected to introduce water into the header 62. It will be noted that the two grids are completely independent of one another in the sense that water or heat transfer fluid introduced into one can not flow to the other, except through the mixing boxes 31, one of which is shown in Fig. 2 with a supply line 59 connected to one of the sprinkler conduits 58 of the first grid, and a return line 60 connected to one of the sprinkler conduits 61 of the second grid.

Referring, again, to Fig. 1, the lines 108 and 109 are operably connected to a line 110, which, in turn, is connected to a source (not illustrated) for water to be used in case of fire. When the apparatus of Fig. 1 is in normal operation, an alarm check valve 111 prevents the flow of water from the line 110 to the lines 108 and 109, and a check valve 112 prevents the flow of heat transfer fluid from the header 57 through the line 108 to either of the

lines 109 and 110. Since heat transfer fluid is supplied to the first grid, entering the header 57, and returns from the second grid, leaving the header 62, the pressure in the header 57 and in the line 108 exceeds that in the header 62 and in the line 109; this pressure difference prevents a flow of heat transfer fluid through the check valve 112 from the line 109 to the line 108.

The apparatus also includes a make up line 113 for heat transfer fluid which flows from a source (not illustrated) through the line 113, a valve 114 and an orifice 115 to the return line 65. The valve 114 is controlled to maintain a constant pressure at the point where the lines 113 and 65 connect. Whenever there is an excessive loss of heat transfer fluid from the system, for example, because a sprinkler head has opened, there is a pressure drop across the orifice 115. This pressure drop is sensed by any suitable means (not illustrated) and the apparatus is put in fire mode by opening the alarm check valve 111 and closing the valve 114 and a valve 116 in the line 63. Opening the alarm check valve 111 enables water for fire purposes to flow from the line 110 into both of the lines 108 and 109. This water is at a pressure sufficiently high that it flows through the check valve 112 even if there is still a reverse pressure from heat transfer fluid on the valve. As a consequence, the water flows through both of the lines 108 and 109, to the first and second sprinkler grids, and through the grids, as required, to the one or ones of the sprinkler heads 107 from which heat transfer fluid had started to flow. Closing the valves 114 and 116 prevents the flow of heat transfer fluid to the coils 48 by stopping the return to the pumps 52.

Another sprinkler system according to the instant invention is indicated generally at 117 in Fig. 3. This system comprises a first grid made up of a loop 118 and sprinkler conduits 119 and 120, and a second grid made up of a loop 121 and sprinkler conduits 122 and 123. The loop 118 is composed of four conduits, 124, 125, 126 and 127 which are operably connected so that a liquid could flow around

the loop 118. The sprinkler conduits 119 are operably connected to the conduit 124 while the sprinkler conduits 120 are operably connected to the conduit 126. The loop 121 is also composed of four conduits, 128, 129, 130 and 131, which are operably connected to one another. The sprinkler conduits 122 are operably connected to the conduit 128, while the sprinkler conduits 123 are operably connected to the conduit 130. There are sprinkler heads 132 in the sprinkler conduits 119, 120, 122 and 123. The heads 132 are spaced from one another a given distance, say 10 feet (3.05 meters) in each of the conduits, and the sprinkler conduits of each grid are spaced from one another a given distance, say 20 feet (3.1 meters). Each of the sprinkler conduits of the first grid is spaced $\frac{1}{2}$ the given distance from one of the sprinkler conduits of the second grid or from two such conduits, depending upon its position in the grid. Accordingly, the two grids jointly constitute a sprinkler system in which conduits are spaced from one another by, say, 10 feet (3.05 meters) and in which sprinkler heads in a given conduit and in an aligned conduit are spaced from one another by, say, 10 feet (3.05 meters). The first grid, however, is connected to the line 108 as a source for water to be used in case of fire, while the second grid is connected to the line 109. Similarly, the line 56 is connected to discharge heat transfer fluid into the first grid, while the line 63 is connected to receive heat transfer fluid from the second grid. The two grids are completely independent of one another in the sense that water or heat transfer fluid introduced into one can not flow to the other, except through the mixing boxes 31, one of which is shown in Fig. 3 with a supply line 59 connected to one of the sprinkler conduits 119 of the first grid, and a return line 60 attached to one of the sprinkler conduits 122 of the second grid.

The apparatus of Fig. 4 includes all of the elements of that of Fig. 1, all designated by the same reference numerals, and, in addition, a waste heat recovery unit 133, absorption refrigeration apparatus indicated generally at

134, pipes 135 and 136, and valves 137, 138, 139 and 140. The unit 133 is operably connected to supply heat to energize the apparatus 134. When the gas engine generator 79 is operating and the apparatus of Fig. 4 is being used on summer cycle to air condition a building, the valves 137 and 138 are open and the valves 139 and 140 are set so that heat transfer fluid discharged from the pumps 52 is directed into the line 136, flows through the absorption apparatus 134, and is cooled before flowing through the line 135 to the main header 55 and from thence, as previously described, through the coils 48 and back to the pumps 52. In this mode of operation, there is no need for the compressor 32 to operate, as the chilled water required for the coils 48 is provided by the absorption refrigeration apparatus 134, supplemented, if required, by heat transfer from the heat transfer fluid in the heat exchanger 90 as previously described. Heat from the absorber and condenser (not illustrated) of the apparatus 134 can be transferred to the cooling tower 82.

20 The apparatus of Fig. 5 includes some of the elements of that of Fig. 1, specifically, the air handler 30, the mixing boxes 31 (one of which is shown in Fig. 5), the sprinkler system comprising the headers 57 and 62 and the sprinkler conduits 58 and 61, the supply and return lines 56 and 63 for circulating chilled water through the sprinkler system to the coils 48 of the mixing boxes 31, and the lines 108 and 109 for supplying water to the sprinkler system when the apparatus is in fire mode. Cooling is provided by compression refrigeration apparatus which includes a compressor 141 and by compression refrigeration apparatus which includes a compressor 142. The compressors 141 and 142 both operate on day cycle; refrigerant from the former flows to a heat exchanger 143, to an evaporator 144, and back to the compressor 141, the flow being through lines 145, 146 and 147. In the heat exchanger 143 heat is transferred from the refrigerant to water that is circulated from a cooling tower 148 through a line 149 to the heat exchanger 143 and, through a line 150, back to the cooling tower 148. The

refrigerant is expanded in the evaporator 144 as required to provide chilled water at, say, 58°F. (14°C.) for circulation as previously described from the main header 55 to the coils 48 of the mixing boxes 31 and back to the main return 64 and 5 the evaporator 144. Refrigerant flows from the compressor 142 through a line 151 to the evaporative condenser 33 and from thence through a line 152 to the coil 40, where it is expanded to maintain the air leaving the air handler 30 and entering the ducts 39 at a temperature of, say, 42°F. 10 (6°C.), returning to the compressor 142 through a line 153.

It will be appreciated that, on summer day cycle, the apparatus of Fig. 5 is identical with that of Fig. 1, insofar as the operation of the air handler 30 and of the mixing boxes 31 is concerned. However, both the compressor 15 141 and the compressor 142 operate when the load is at a peak; as a consequence, the peak load and the demand charge associated therewith are nearly as high as with the previously described existing equipment, the only energy saving being that attributable to the lesser quantity of 20 colder air that is required to be circulated. It has been determined, however, that the first cost is less than 75 percent of that of duplicating the existing equipment, the lowered cost being attributable to the savings in ductwork and air moving apparatus (fans and motors) which were 25 possible because of the reduced volume of conditioned air to be circulated. As is subsequently explained in more detail with reference to Fig. 12, the energy efficiency of the apparatus of Fig. 5 can be improved significantly by modifying the air handler 30, specifically, by adding a 30 second coil and transferring heat from a mixture of return air and outside air to 58°F. (14°C.) water circulated through that coil from the evaporator 144. This is true because the energy requirement is about 0.5 kilowatt per ton of refrigeration to produce water at 58°F. (14°C.) but about 35 0.85 kilowatt per ton to cool air to 42°F. (6°C.). Accordingly, shifting one third of the load in the air handler 30 to 58°F. (14°C.) water effects about a 15% energy savings for refrigeration therein. Apparatus which includes

such a modified air handler, but is otherwise substantially identical to the Fig. 5 apparatus, is shown in Fig. 12 and described subsequently with reference thereto.

It is possible, in the apparatus of Figs. 1, 4 and 5, 5 as well as in that of Fig. 12, to use the sprinkler system to provide a chilled plenum, or to perform localized cooling. All that is necessary is to connect the sprinkler conduits 58 and 61, or some of them, by lines which contain control valves (not illustrated in Figs. 1, 4, 5 and 12). 10 When the valves are closed, the apparatus delivers chilled water only to the mixing boxes 31, as previously described. However, chilled water flows through each line when the appropriate valve is open and, in addition, through the associated sprinkler conduits 58 and 61. This flow of 15 chilled water is operable to transfer heat from a plenum which contains the sprinkler system, or from selected portions thereof.

Sprinkler apparatus which includes such lines and valves is indicated generally at 154 in Fig. 6. The 20 sprinkler system 154 includes headers 155 and 156 and sprinkler conduits 157 and 158 operatively associated, respectively, with the headers 155 and 156. Conduits 159 connect adjacent ones of the sprinkler conduits 157 and 158. Valves 160 control the flow of heat transfer fluid through 25 the conduits 159. If desired, the valves 160 can be controlled by thermostat controllers, or by a single thermostat controller (not illustrated), so that the flow of heat transfer fluid through the conduits 159 is modulated, as required, to maintain a desired temperature in the 30 associated plenum. For example, the valves 160 can be modulated to maintain the plenum at a temperature of 60°F. (16° C.); heat will then be transferred to the plenum from the adjacent space, thereby reducing the load that must be carried by the coils 48 (Figs. 1, 4, 5, and 12) in the 35 mixing boxes 31 and by the coil 40 in the air handler 308. Fins (not illustrated) can be added to the sprinkler conduits 157 and 158 (Fig. 6) and to the conduits 159 to increase heat transfer from the plenum, if desired. Heat

transfer fluid is delivered to the apparatus 154 from a supply line 161 and leaves through a return 162, while water for fire purposes can be supplied to both of the headers 155 and 156 through conduits 163.

5 Apparatus indicated generally at 164 in Fig. 7 is also a sprinkler system, comprising headers 165 and 166 and operatively associated sprinkler conduits 167 and 168, a mixing box 31 operatively connected between one of the conduits 167 and one of the conduits 168 and, in addition,
10 simplified mixing boxes 169 and cooled lights 170, both of which are served with chilled water from adjacent ones of the sprinkler conduits 167 and 168. The chilled water flows from the sprinkler conduits 167 through conduits 171 to the mixing boxes 169 and to the lights 170 and then through
15 conduits 480 to the sprinkler conduits 168, under the control of valves 173. The apparatus 164 also includes a supply line 174 and a return 175 for a heat transfer fluid, and conduits 176 through which water for fire purposes can be supplied to the headers 165 and 166.

20 One of the mixing boxes 169 is shown in more detail in Fig. 8, mounted so that its bottom is flush with a ceiling 177 of a space it serves and a housing 178 in which a two speed electric motor 179 is mounted extends above the ceiling in a plenum 180. One of the conduits 172 which
25 carries fins 181, extends through a collar 182 which is a part of the housing 178. When the motor 179 is energized, a fan 183 discharges air downwardly from the housing 178 and induces a flow of air into the housing. The induced air can flow (1) directly from the space through an opening 184 in
30 the ceiling and an opening 185 in the wall of the housing 178, (2) directly from the plenum through the collar 182 which extends upwardly from the top of the housing 178, or (3) partially from the space through an opening 186 in the ceiling 177 and partially from the plenum into one or both
35 of two openings 187 and 188, depending on the positions of dampers 189 and 190.

In operation, the mixing box 169 discharges air into the space it serves at a rate which depends upon the speed

of the motor 179. Some of this air is induced to flow from the space through the openings 184 and 185; some is induced to flow from the plenum through the collar 182; and some may be induced to flow through one or both of the openings 187 and 188, depending on the positions of the dampers 189 and 190. The mixing box 31 (Fig. 7) is used as previously described to introduce conditioned 42°F. (6°C.) air into a zone served by the mixing box 31 and a plurality of the mixing boxes 169, the rate of delivery of the conditioned air being sufficient to provide humidity control for the entire zone. Each of the motors 179 (Fig. 8) of the mixing boxes 169 can operate at high speed or at low speed, or can be de-energized, depending on the air conditioning load in the zone and whether or not the zone or a portion thereof is occupied. For example, if the apparatus of Fig. 7 serves a multi-story department store, the mixing boxes 31 can be energized at the start of a business day, and operated to deliver a mixture of 42°F. (6°C.) primary air and recirculated air as previously described, and motors 179 (Fig. 8) of the mixing boxes 169 can remain de-energized until a motion sensor (not illustrated) indicates that the zone they serve is occupied. The motion sensor, which can be, for example, of the type disclosed in U.S. patent No. 4,485,864, can be operably connected to energize the motors 179 of the mixing boxes 169 which serve the space in which motion has been sensed, and to continue them in an energized condition so long as motion continues to be sensed. While the motors 179 are energized, the operation of the mixing boxes 169 and of the valves 173 (Fig. 7) can be controlled by a thermostatically operated controller (not illustrated). If a temperature substantially higher than the control temperature is sensed in an occupied zone, some or all of the valves 173 which serve that zone can be opened to enable a flow of heat transfer fluid at 58°F. (14°C.) through the conduits 171 and 172 which serve the relevant ones of the mixing boxes 169 or which serve the relevant portion of the entire sprinkler system. If all of the valves 173 in the relevant portion of the sprinkler system are opened, that

portion of the plenum will be cooled, and then, maximum cooling from the mixing boxes 169 can be achieved by inducing a maximum flow of air into the mixing boxes 169 from the plenum. If only the valves 173 which control the flow of heat transfer fluid through those of the conduits 171 and 172 which serve the relevant mixing boxes 169 are opened, maximum cooling can be achieved by inducing a maximum flow of air into the mixing boxes 169 in heat transfer relationship with the ones of the conduits 171, 172, or both through which the heat transfer fluid is flowing. As the sensed temperature approaches the control temperature, all of the relevant ones of the valves 173 which were opened, can be modulated or closed, the dampers 189 and 190 (Fig. 8) can be controlled to reduce heat transfer to the circulated fluid, or both. In general, the maximum flow of induced air from the plenum 180 occurs when the damper 189 is open and the damper 190 is closed, while the maximum flow of induced air through the collar 182 occurs when both of the dampers 189 and 182 are closed. The temperature of the plenum 180 will be a function of heat gains attributable to the lights 170, to other lights and electronic equipment (not illustrated) in the space, and the like and of heat losses to fluid circulated through the sprinkler system 164 (Fig. 7). Accordingly, the apparatus 164 can be controlled so that heat is transferred to the space therefrom, or so that heat is transferred thereto from the space, and the amount of heat so transferred can be controlled in a simple manner to maintain a desired space temperature. The amount of heat so transferred can also be changed by changing the speed of the motors 179, the maximum transfer being accomplished at the high speed, and a lesser transfer at the low speed.

In general, the flow of heat transfer fluid through the sprinkler apparatus 164 can be controlled to maintain the entire plenum 180 (Fig. 8) or the portion thereof in the vicinity of any one or any desired group of the mixing boxes 169 at a desired temperature, which can range from 60°F. (16°C.) to 80°F. (27°C.). The flow of conditioned air from

the mixing boxes 31, as is subsequently explained in detail, can be controlled by a humidistat/controller (not illustrated in Figs. 7 and 8) to effect humidity control in the space and, because there is a flow of air thereinto from the space, in the plenum. Humidity control is necessary in the plenum to prevent condensation on the heat transfer devices which are used as described above to control temperature, including the headers, sprinkler conduits and other conduits through which a heat transfer fluid at, say, 58°F. (14°C.) may be circulated. The mixing boxes 169 which serve a given zone of the space can be controlled together, in response to signals from a single thermostat, or can be so controlled individually to provide several different temperature zones in the zone for which a single mixing box 31 provides humidity control.

The light 170 (Fig. 7) is shown in more detail in Fig. 9, comprising a high intensity bulb 191 received in a socket 192 which is mounted in a housing 193. A sheet 194 of metal fabric is draped over the conduit 172 and extends downwardly on both sides of the bulb 191 where it is heated by thermal energy it intercepts and also by radiant energy. Because the sheet 194 is a fair conductor of heat and is in thermal contact with the conduit 172, heat is transferred therefrom to the conduit 172 and to heat transfer fluid circulated through the conduit 172, thus minimizing the storage of heat from the bulb 191 in the building structure and ultimate release of the stored heat at times of maximum air conditioning load.

Lighting fixtures which are disclosed in U.S. patent No. 3,828,180 and other water-cooled lighting fixtures can also be connected between the conduits 167 and 168, and the flow of heat transfer fluid therethrough can be controlled so that lighting heat is either used for reheat or transferred to a major extent to the heat transfer fluid. For example, whenever the air conditioning load on a space served by one or a plurality of the mixing boxes 31 is such that heating is required, the flow of heat transfer fluid to the lighting fixtures that serve that zone can be modulated

so that as much of the lighting heat as is required is available to the space. Specifically, the lighting fixtures of the '180 patent have dampers which can be opened when lighting heat is required in the space; when the dampers are 5 open, air can flow from the space into the lighting fixtures and through openings in the fixtures into a plenum from which the mixing boxes 31 induce a flow of air. Such flow through the fixtures is prevented when the dampers are closed. Lighting fixtures of this type can also be used in 10 conjunction with the apparatus of Figs. 15-17 which is subsequently described in detail.

A particularly advantageous control device for the mixing box 31 of Figs. 1, 4, 5 and 12 is shown, somewhat schematically, in Fig. 10. The device comprises a controller 15 195 for the damper 45 and for the valve 66. Signals from a humidistat or thermostat 196, from a humidistat 197 and from a thermostat 198 are input to the controller 195 which then controls the damper 45 and the valve 66. The humidistat 197 and the thermostat 198 sense conditions in a space 199 20 served by the mixing box 31, while the humidistat or thermostat 196 senses conditions inside the mixing box 31, specifically of air that has been induced to flow from the space 199 into the mixing box 31 and has been cooled by heat exchange with the coil 48. This combination of sensors with 25 the controller 195 is well suited for use when different spaces served by different ones of the mixing boxes 31 are to be maintained at different humidities or have substantially different humidity loads. It is important to prevent condensation, which occurs whenever air is cooled to 30 a relative humidity of 70 % or higher, on the coils 48. The controller 195 prevents such condensation; when the mixing box 31 is first energized, the controller closes the valve 66, keeps it closed until a signal received from the thermostat or humidistat 196 indicates a relative humidity 35 below 70% and thereafter controls the valve 66, if necessary, to keep the relative humidity below 70%, its function in this mode being in the nature of a limit switch. The controller 195 also opens or closes the damper 45,

initially, when the mixing box 31 is first energized, as required to establish and maintain the control temperature as sensed by the thermostat 198, and thereafter as required to establish and maintain the control humidity as sensed by the humidistat 197. The controller 195, when it is in the latter mode, controlling the damper 45 to maintain the control humidity as sensed by the humidistat 197, also controls the valve 66 to maintain the control temperature as sensed by the thermostat 198, the limit on opening of the valve being that position at which the signal from the humidistat or thermostat indicates a relative humidity of 70%, as described above.

The signal from the humidistat or thermostat 196 is a direct indication of relative humidity only when the instrument is a humidistat; the signal must be compared with the signal from the humidistat 197 for the required indication of relative humidity when the instrument is a thermostat.

The controller 195 can also operate in another manner, modulating the damper 45, when the air conditioning load in the space is comparatively light, to maintain the absolute humidity sensed by the humidistat 197 at a set point, say, 64 grains of water vapor per pound of dry air, and

(1) so long as the humidistat 196 senses a relative humidity not greater than, say, 70 percent, modulating the valve 66 to maintain the space temperature sensed by the thermostat 198 at a set point, say, 78°F. (26°C.), or

(2) so long as the humidistat 196 senses a relative humidity greater than, say, 70 percent, closing the valve 66, if open, or keeping the valve 66 closed, if it is already closed, and opening the damper 45 to reduce the relative humidity. This mode of operation also prevents condensation on the coil 48.

When the apparatus of Fig. 10 is operated in either of the modes described above and the air conditioning load is sufficiently high that the valve 66 is in its fully open position and the temperature sensed by the thermostat 198 is above the current set point, the set point for the

humidistat 197 can be lowered and the set point for the thermostat 198 can be raised. If the valve 66 again reaches a full open position, the set points can be reset again. Other suitable set points are:

5	<u>Humidstat 197</u>	<u>Thermostat 198</u>
	60*	80°F. (27°C.)
	56*	82°F. (28°C.)
	<u>52*</u>	84°F. (29°C.)

*Grains of water vapor per pound of

10 dry air.

Ordinarily, it is desirable for apparatus according to the instant invention to be designed so that changing the set point for the humidistat 197 is necessary only when the air conditioning load is heavy. This is true because the use of
 15 primary conditioned air at 42°F. (6°C.) is a considerably more expensive way to counteract sensible heat gains than is the use of 58°F. (14°C.) water in the coil 48. Consequently, the controller 195 reverts to the next higher humidity set point for the humidistat 197 and the corresponding set point
 20 for the thermostat 198 after it has operated for, say, thirty minutes at any given reduced humidity set point. If the thermostat 198 senses too high a temperature with the valve 66 in its full open position, the lowered humidity and increased temperature set points will be reinstated, as
 25 described above, and will remain in effect for another short period of time, say, thirty minutes, unless, in the meantime, the thermostat 198 senses too high a temperature with the valve 66 in its full open position. When the load decreases again, the original set points will be reinstated
 30 in the manner just described.

It is also possible for the controller 195 to revert to the next lower humidity set point for the humidistat 196 whenever the valve 66 is throttled to its closed position or to any desired position between full open and closed, but
 35 reversion on the basis of elapsed time is preferred. A humidistat/thermostat controller which can be programmed to operate in any of the ways the controller 195 is described

herein as operating can be purchased from VAISALA, Inc., Woburn, Maine.

In installations where approximately the same relative humidity, say 50 %, is to be maintained in all of the spaces 5 served by air conditioning apparatus according to the instant invention, provided that there are not large fluctuations in humidity load, simpler control apparatus than that shown in Fig. 10 can be used for summer operation. Specifically, the humidistat or thermostat 196 and the 10 humidistat 197 can be eliminated, and the signal from a humidistat 200 (Figs. 1, 4, 5 and 12) can be used to control each of the mixing boxes 31 as described below. The humidistat 200 senses the absolute humidity of return air from all of the spaces served as that air flows through a 15 duct 201. On start-up of the apparatus, a signal from the humidistat 200 is input to a controller 202 for the pumps 52, which are energized only when that signal indicates that the absolute humidity of the return air in the duct 201 is at or below a set point, say 64 grains of water vapor per 20 pound of dry air. A signal from the humidistat 200 is also input to each of the controllers 195 (Fig. 10), as is a signal from the thermostat 198 associated therewith. Each of the controllers 195 operates the damper 45 associated with it to maintain a set temperature, say 78°F. (26°C.), in the 25 space it serves and opens the valve 66 associated with it whenever the relative humidity of recirculated room air in the mixing box 31, at the lowest temperature to which the coil 48 is capable of cooling it and at the absolute humidity sensed by the controller 195 (Figs. 1, 4, 5 and 30 12), is less than 70%. When the apparatus is controlled as just described, only air from the duct 39 is available to lower the temperature of the space served by each of the mixing boxes 31 during the first part of morning start-up. Since this air has a low humidity, being saturated at 42°F. 35 (6°C.), each of the spaces is also dehumidified even though the dampers 45 are controlled only on the basis of dry bulb temperature. However, as soon as the humidistat 200 senses a humidity sufficiently low to indicate that humidity control

has been established, the pumps 52 are energized to circulate chilled water from the water chiller 35, and the valves 66 are opened to make that chilled water available to remove heat from the space served by each of the mixing boxes 31. As soon as a signal from one of the thermostats 198 (Fig. 10) indicates that the space served by the associated one of the mixing boxes 31 has reached the set temperature, the controller 195 throttles and then modulates the damper 45 (Figs. 1, 4, 5 and 12) to maintain the set temperature. The air handler 30 is operated to maintain a predetermined static pressure at a point in the duct 39; accordingly, when the dampers 45 are throttled, as just described, the rate at which conditioned air is delivered to the duct 39 and the rate at which ice is used to produce the conditioned air are both reduced. The valves 66 remain in their fully open positions unless the temperature in a space served by one of them is below the set point with the relevant damper closed to the point where only the minimum ventilation air is being supplied; that one of the valves 66 is then modulated by the controller 195 for temperature control and, if necessary, a resistance heater 203 is energized. A heat pump (not illustrated) can also be added to the mixing boxes 31 to pump heat from the heat transfer fluid circulated through the sprinkler system (see Figs. 1, 4, 5 and 12) to a coil (not illustrated) in the mixing box 31 in heat transfer relationship with air circulated therethrough; preferably, the coil (not illustrated) is in heat transfer relationship with the mixture of recirculated air and conditioned air so that, when the air conditioning load is high, heat can be pumped from the air to the heat transfer fluid with only a minimal risk of condensation because the mixture has a low humidity.

The mixing box 31, equipped with the controller 195 of Fig. 10, is shown in Fig. 11 with a heat exchanger 204 added. Primary air from the duct 39 (at 42°F., 6°C.) enters one side of the heat exchanger 204, leaving through a duct 205 from which it enters the mixing box 31 where it is mixed with recirculated air from the space. The mixture of

recirculated air from the space and of primary air from the duct 205 flows in thermal contact with the coil 48 and then through a second side of the heat exchanger 204. Heat is transferred, in the heat exchanger 204, from the mixture of 5 primary air and recirculated air to incoming primary air, and the air which flows in heat transfer relationship with the coil 48 is cooler and drier, other factors being equal, than in the mixing boxes 31 of Figs. 1, 4, 5, 10 and 12; as a consequence, the likelihood of condensation on the coil 48 10 is reduced.

The apparatus of Fig. 12 includes all of the elements of that of Fig. 5, as indicated by the use of the same reference numerals, except that the air handler 30 has been replaced by an air handler 206 in the Fig. 12 apparatus. The 15 air handler 206 has the coil 40 which cools air circulated thereover to 42°F. (6°C.) and, in addition, has a coil 207 to which water at, say, 58°F. (14°C.) is circulated through lines 208 and 209. The load that must be carried by the coil 40 is reduced as a consequence of heat transfer to the coil 20 207; as is explained above, shifting load from the coil 40 to the coil 207 saves energy because only about 0.5 kilowatt per ton of refrigeration is required to produce 58°F. (14°C.) water instead of about 0.85 kilowatt per ton of refrigeration to cool air to 42°F. (6°C.).

25 Another sprinkler system according to the invention is indicated generally at 210 in Fig. 13. The sprinkler system 210 comprises a water chiller 211, a water heater 212, fan coil mixing boxes 213, sprinkler conduits 214, sprinkler heads 215, make-up apparatus indicated generally at 216 to 30 maintain a constant heat transfer fluid pressure in the system in normal operation, means indicated generally at 217 for introducing water for fire purposes into the apparatus, a pump 218 and piping, valves, orifices and the like for circulating a heat transfer fluid to the mixing boxes 213 in 35 normal operation and water for fire purposes to the affected ones of the sprinkler heads 215 in fire mode.

The apparatus 216 serves a multi-story building; enough of the portion thereof for one floor to explain the

operation is shown, enclosed within a broken line, together with fragments thereof for other floors. In normal operation, which is the same for all floors, the pump 218 causes a heat transfer fluid to flow through a supply line 5 219, valves 220, orifices 221, headers 222, conduits 223, supply pipes 224, the mixing boxes 213, return pipes 225, orifices 226, conduits 227, return headers 228, pipes 229 with check valves 230 therein, a return line 231, a pipe 232 and the heater 212 or the chiller 211 back to the pump 218. 10 Check valves 233 prevent the flow of heat transfer fluid from the headers 222 through lines 234 to the return headers 228, forcing the flow, instead, through the mixing boxes 213, as described.

The make-up apparatus 216 comprises a pressurized tank 15 235 which is connected by a pipe 236 to the return line 231. In normal operation, the pressure of the tank 235 is imposed on the heat transfer fluid in the return line 231 and there will be a minor flow of fluid from the tank 235 to the return line 231 or vice versa to accomodate minor losses of 20 heat transfer fluid from the apparatus, expansion and contraction of the heat transfer fluid in the apparatus, and the like. If one of the sprinkler heads 215 is fused by a fire, heat transfer fluid flows from that head 215 at a comparatively rapid rate and, as make-up, from the tank 235 25 through the pipe 236 and through a restricting orifice 237 therein. A pressure drop across the orifice 237 is sensed by a sensor-controller (not illustrated), which then puts the apparatus 210 in fire mode by closing the valves 220, de-energizing the pump 218, energizing a fire pump 238 and 30 setting valves in the means 217 so that fire water from a main 239 is delivered to the return line 231 from which it flows through one of the return headers 228, through a check valve 240 therein, and to the fused head or heads 215. The check valves 233 do not interfere with the flow of fire 35 water from the side of the return line 231 through the lines 234. Water for fire purposes can also be introduced into the apparatus 210 through a siamese 241 or through a siamese 242.

Referring to Fig. 14, which is a schematic diagram of the apparatus of Fig. 4 showing features which were omitted from Fig. 4, and from which certain features have been omitted to facilitate the showing of the new features, heat from both the exhaust gases and the jacket water from the engine-generator 79 can be transferred to the absorption refrigeration apparatus 134. The exhaust gases are discharged from the engine generator 79 through a stack 243, and are either vented through a discharge 244 or directed through a branch line 245 into heat exchange relationship with the absorption refrigeration apparatus 134, depending upon the setting of a damper 246. Exhaust gasses which are directed through the branch line 245, after having furnished heat to energize the apparatus 134, flow into and are vented from a stack 247. Jacket water leaves the engine-generator 79 through a line 248, and is directed into a line 249 or into a line 250, depending on the setting of a valve 251. On summer cycle when chilled water from the absorption apparatus 134 is required, the water is circulated through the line 250, a heat exchanger 252, a line 253, a heat exchange jacket 254 of the apparatus 134, a line 255, the heat exchanger 252 and a line 256 back to the engine-generator 79, providing energy for the apparatus 134. Whenever heat is required in the air handler 30, the valve 251 is set so that jacket water from the engine-generator 79 is circulated through the line 249, a sheet and tube heat exchanger 257, and a line 258 to the line 250, and then, as previously described, through the heat exchanger 252, the heat exchange jacket 254, the heat exchanger 252 and back to the engine-generator 79. A valve 259 is modulated, as required, to maintain a desired temperature; in one position, the valve 259 causes circulated jacket water to flow through the heat exchanger 257 while, in the other, it causes water to flow through a line 260, bypassing the exchanger 257. Jacket water flowing through the exchanger 257 heats a heat transfer fluid, for example ethylene glycol, which is circulated from the heat exchanger 257

through a line 261, a coil 262 in the air handler 30, and a line 263 back to the exchanger 257.

Apparatus shown in Fig. 15 is similar to that of Fig. 1, the main differences being that: the air handler 30 is not included in the Fig. 15 apparatus; dehumidifiers 264 and 265 have been added to perform the function of dehumidifying air; and, on day cycle, ice from the ice storage tank 34 can be used to provide chilled water for mixing boxes 266, one of which is shown in Fig. 15. The following elements of the Fig. 1 apparatus are included: the compressor 32, the evaporative condenser 33, the ice storage tank 34, the water chiller 35, the high pressure receiver 69, the low pressure receiver 71 and the heat recovery unit 96, all of which perform substantially the same as in the Fig. 1 apparatus, except as described below.

Ambient air is introduced into the apparatus of Fig. 15 by a blower 267, flowing through a filter 268, a duct 269 and the dehumidifier 264 and then into the blower 267, which discharges into a duct 270 from which the air flows through the dehumidifier 265, and then through a duct 271 to the mixing boxes 266.

The apparatus also includes a cooling tower 272 from which evaporatively cooled water is circulated through a line 273 to a heat exchanger 274 and then through a line 275 back to the cooling tower 272. Chilled water from the water chiller 35 is circulated through lines 276, 277 and 278 to a heat exchanger 279 and then through lines 280, 281 and 282 back to the water chiller 35.

In operation, a desiccant, e.g., an aqueous lithium chloride solution, is circulated from a sump 283, through a line 284, the heat exchanger 274 and a line 285 from which it is sprayed in the dehumidifier 264 in contact with air flowing therethrough, returning by gravity through a line 286 to the sump 283. Similarly, the desiccant is circulated from the sump 283 through a line 287, the heat exchanger 279 and a line 288 from which it is sprayed in the dehumidifier 265 in contact with air flowing therethrough, returning by gravity through a line 289 to the line 287.

Desiccant in the sump 283 is maintained at a predetermined concentration, say 40 to 42 weight percent lithium chloride, by a regenerator 290 to which desiccant flows from the sump 283 through a line 291, a heat exchanger 5 292 and a line 293, and from which concentrated desiccant is returned to the sump 283, flowing through a line 294, the heat exchanger 292 and a line 295.

Typically, the dehumidifiers 264 and 265 can operate to condition ambient air, which may have a dry bulb temperature 10 of 91°F. (33°C.) and a moisture content of 124 grains of water vapor per pound of dry air so that the air in the duct 269 has a dry bulb temperature of 94°F. (34°C.) and a moisture content of 42 grains of water vapor per pound of dry air, while the air in the duct 271 has a dry bulb 15 temperature of 75°F. (24°C.) and a moisture content of 31 grains of water vapor per pound of dry air. A space condition of, say, 76°F. (24°C.) dry bulb temperature, 50 percent relative humidity (67 grains of water vapor per pound of dry air), is maintained by controlling the rate at 20 which air from the duct 271 is delivered to each of the spaces served by the apparatus (one space, designated 296, is shown in Fig. 15) to maintain humidity, and by controlling the temperature, as subsequently explained, at which a mixture of air from the duct 271 and recirculated 25 air is delivered to the space. A thermostat-humidistat controller 297 controls a damper 298 to vary the rate at which air from the duct 271 is delivered to each of the spaces 296. The rate may vary between 0.1 and 0.2, or even 0.3 cubic foot per minute per square foot of floor space in 30 a given zone; the controller 297 opens the damper 298 incrementally whenever the humidity is too high and closes it incrementally whenever the rate is above the minimum required for ventilation and the humidity is too low.

Each of the mixing boxes 266 includes a blower 299 35 which induces a flow of air from a plenum, as indicated by an arrow 300, and delivers to the space 296 it serves a mixture of that induced air and conditioned air from the duct 271. The flow of induced air from the plenum into the

mixing box 266 causes a flow of air from the space 296 into the plenum; two arrows 301 indicate the delivery to the space 296 of a mixture of induced air and conditioned air and the flow of air from the space 296 into the plenum. 5 Inside the mixing box 266, the mixture of conditioned air and induced air flows in heat exchange relationship with a coil 302. When the apparatus is in cooling mode, chilled water is circulated, as subsequently described in detail, to a sprinkler conduit 303. This water flows through a line 10 304, a control valve 305, the coil 302 and a line 306 back to a sprinkler conduit 307. The controller 297 also modulates the valve 305, closing it incrementally whenever the space temperature is below the control temperature, and opening it incrementally whenever the temperature is above.

15 A sprinkler main 308 is connected to the line 277 and to the sprinkler conduit 303, while a sprinkler main 309 is connected to the line 281 and to the sprinkler conduit 307. As a consequence, chilled water at, say, 58° F. (14°C.) from the water chiller 35 is delivered to the coils 302 for 20 temperature control. Water in the line 281 returning from the coils 302 and from the heat exchanger 279 can be diverted by a valve 310 so that it flows through a line 311 to a heat exchanger 312, returning through a line 313 to a heat exchanger 314, and through a line 315 and the valve 310 25 to the line 282. Heat can be transferred in the heat exchanger 312, when conditions are appropriate, to evaporatively cooled water from the cooling tower 271, and more heat can be transferred in the heat exchanger 314 to ice in the ice storage tank 34. Accordingly, the compressor 30 32 can be operated on summer night cycle to produce ice that is used on summer day cycle to carry all or any part of the sensible load on the coils 302 and all or any part of the load on the heat exchanger 279.

The apparatus of Fig. 15 also includes an engine 35 generator 316 which furnishes electricity as indicated at 317 to the pumps, blowers and the like of the apparatus, to the electrical service of the building it serves, or both. Heat from a cooling jacket (not illustrated) of the

engine-generator 316 is transferred to water circulated by a pump 318 through a line 319, a heat exchanger 320, lines 321 and 320, a heat exchanger 323 and a line 324 back to the cooling jacket of the engine-generator 316. In the heat exchanger 320, heat is transferred from exhaust gases from the engine-generator 316 to water circulated therethrough. In the heat exchanger 323, heat is transferred from the water circulated by the pump 318 to a heat transfer fluid that is circulated to a heat exchanger 325 and, on winter cycle, to a heat exchanger 326, and to a heating coil 327.

On summer day cycle, a part of the desiccant solution flowing in the line 294 is diverted, flowing through a line 328, the heat exchanger 325 and a line 329 to spray nozzles 330 from which it is sprayed in the regenerator 290 to remove water from desiccant therein, so that highly concentrated desiccant solution flows from the bottom of the regenerator 290 through a line 331 to a sump 332. It is to the sump 332 that dilute desiccant flows through the line 293, and from the sump 332 that a pump 333 delivers relatively concentrated desiccant solution to the line 294, as described. Relief air from the building served by the apparatus is delivered through a duct 334 to a blower 335, from which it is discharged through a heat exchanger 336 into the regenerator 290, leaving through a duct 337 and a heat exchanger 338. A heat transfer fluid is pumped from the heat exchanger 338 through a line 339 to the heat exchanger 336 and through a line 340 back to the heat exchanger 338 to recover heat from air leaving the regenerator 290 and to transfer that heat to air entering the regenerator 290.

The concentrated desiccant returned through the line 294 to the sump 283 may have a concentration of 42 percent by weight of lithium chloride, and may maintain a concentration of 40 percent by weight in the sump 283. A pump 341 delivers desiccant from the sump 283 directly to the line 284, so the concentration of the desiccant sprayed in the dehumidifier 264 is also 40 percent by weight. A pump 342 receives desiccant from the line 287, but this desiccant is a mixture of desiccant from the sump 283 and more dilute

desiccant from the dehumidifier 265. As a consequence, the desiccant delivered to the line 288 may contain 38 percent by weight of lithium chloride; a part of this desiccant is sprayed in the dehumidifier 265, as previously described, 5 while the rest is returned to the sump 283, flowing through a line 343 and the line 286.

Any excess heat from the engine generator, beyond that used by the apparatus of Fig. 15, as described above, can be rejected through a roof-mounted fan radiator 344 to which 10 water circulated by the pump 318 can flow through a line 345, returning through a line 346. The rejection of heat in this manner can be controlled by a valve 347 which is modulated, as required, to prevent the temperature of the water from becoming excessively high.

15 It will be appreciated that the conditioned air delivered to the mixing boxes 266 when the Fig. 15 apparatus is operated as described above is essentially neutral air, so far as temperature control is concerned. That is, the air temperature is about the same as that to be maintained in 20 the spaces it serves. Accordingly, when the rate at which conditioned air is delivered to any given one of the spaces is varied because of changes in the humidity load, the variations do not increase or decrease the sensible load that must be transferred to heat transfer fluid from 25 circulated air in the coil or coils 302. The reason for this is that all of the air that enters the mixing boxes 266 is neutral so far as temperature control is concerned; when the rate at which conditioned air is delivered to any given one of the mixing boxes 266 increases, the rate at which plenum 30 air is delivered thereto decreases correspondingly, so that the total flow of air into the mixing box 266 remains constant. The apparatus of Fig. 15 is significantly different, in this respect, from that of Figs. 1, 4, 5 and 14, where the conditioned air is both cold and dry, so that 35 a change in the rate at which it is delivered to a given space changes the sensible load that must be carried by heat exchange from circulated air (which is recirculated air in those embodiments of the invention). However, as is

subsequently explained in more detail, the apparatus of Fig. 15, as well as that of Fig. 17, can be operated so that the temperature of the dehumidified air which is circulated through the duct 271 to the mixing boxes 266 is above the temperature maintained in the spaces 296, for example, from the temperature maintained up to about 90°F. (32°C.).

Because the conditioned air delivered by the apparatus of Fig. 15 is relatively warm, insulated ducts are not required for the circulation thereof; the air is not capable of causing condensation. Indeed, the conditioned air can be delivered from a riser of a multistory building to a header on each floor, and can then be circulated as required for a given floor through a cellular deck, for example of the type shown in U.S. patents No. 3,013,397 and No. 3,148,727.

The mixing box 266 of the Fig. 15 apparatus also includes a unitary heat pump 348 which has a heat exchange coil (not separately illustrated) between the coil 302 and the blower 299, a compressor (not separately illustrated) and a condenser or evaporator (not separately illustrated). Two lines 349 operably connect the condenser or evaporator of the heat pump 348 to the lines 304 and 306; when a valve 350 is opened by the controller 297, water flowing through the lines 349 constitutes a heat source or a heat sink for the condenser or evaporator of the heat pump 348, returning to the line 306 after it has served its purpose in the heat pump 348. There are times when the cooling tower 272 provides all of the cooling that is required by most zones of the building served by the apparatus of Fig. 15, but is not quite adequate for a few of the zones. At these times, water from which heat has been transferred by the cooling tower 272 can be circulated to the mixing boxes 266, and, under the control of the thermostat-humidistat controllers 297, the heat pumps 348 can be energized to pump heat from the heat exchange coils of the heat pumps 348 which serve the zones where additional cooling is required. When such additional cooling is required, it is important to prevent condensation on the evaporator of the heat pump 348. To that end, it is desirable that the thermostat-humidistat

controller 397 include a humidistat (not illustrated) which senses the humidity of the air between the heat pump 348 and the blower 299, and de-energizes the heat pump 348, opens the damper 298, or both, whenever that relative humidity exceeds 70 percent.

When the compressor 32 is operating to produce ice in the storage tank 34 and heat is required by the building served by the apparatus of Fig. 15, water that has been heated by heat from the heat recovery unit 96 can be circulated through a line 351 to the line 277 and then, as described above, to the coils 302, returning to the heat recovery unit from the line 281 through a line 352. In this mode of operation, a valve 353 is positioned so that all of the fluid returning in the line 281 is directed into the line 352; as a consequence, there is no fluid flow in the line 311 or in the line 282, and there is no heat transfer from the circulating system to the water chiller 35, to the storage tank 34 or to the cooling tower 272. The heat pumps 348, in this mode of operation, can be operated to pump heat from the fluid circulated thereto through the lines 349 if additional heating is required in the zones served by some of the mixing boxes 266.

The apparatus of Fig. 16 is identical in most respects with that of Fig. 15, the principal difference being that the Fig. 15 dehumidifier 264 has been replaced, in the Fig. 16 apparatus, by a solid desiccant dehumidifier indicated generally at 354. The dehumidifier 354 comprises two desiccant wheels 355 and 356, two blowers 357 and 358, and heat exchangers 359, 360, 361 and 362. In operation, the desiccant wheels 355 and 356 rotate slowly, while air to be dehumidified enters a conduit 363, flows through a segment of the desiccant wheel 355, through a conduit 364, through a segment of the desiccant wheel 356, through a conduit 365 into the blower 357 and into the conduit 270. The air is dehumidified when it flows through each of the wheels 355 and 356, as just described, by contact with a solid desiccant, e.g., activated alumina, silica gel or lithium chloride on a paper carrier. For example, ambient air may

enter the conduit 363 at a dry bulb temperature of 93°F. (34°C.) and a moisture content of 105 grains of water vapor per pound of dry air, leave the desiccant wheel 356 at a dry bulb temperature of 104°F. (40°C.) and a moisture content of 56 grains of water vapor per pound of dry air, be cooled by the heat exchanger 359 to a dry bulb temperature of 91°F. (33°C.) with no change in moisture content, and leave the dehumidifier 265 at a dry bulb temperature of 75°F. (24° C.) and a moisture content of 31 grains of water vapor per pound of dry air. In the heat exchanger 359 heat is transferred from the dehumidified air to evaporatively cooled water which is circulated from the cooling tower 272 through the line 273 to the heat exchanger 359 and through the line 275 back to the cooling tower 272.

Relief air from the building served by the dehumidifier 354 is used for regeneration, entering the blower 358 from a conduit 366, flowing through a segment of the desiccant wheel 355, through a conduit 367, through a segment of the desiccant wheel 356, and through a conduit 368 from which it is vented to the atmosphere. A heat transfer fluid is circulated by a pump 369 from the heat exchanger 362 to the heat exchanger 361 and back, the flow being through lines 370 and 371; as a consequence, heat is transferred from air leaving the regeneration side of the dehumidifier to air that is about to flow in regenerating relationship with the desiccant of the wheel 356. In addition, heat transfer fluid is circulated through the heat exchanger 361, flowing thereto from the heat exchanger 323 through lines 372 and 373, and returning to the heat exchanger 323 through lines 374 and 375. In this way, heated water circulated by the pump 318 furnishes the heat required for the regeneration of the wheel 356.

The apparatus of Fig. 17 is similar to that of Fig. 15, differing in that the compressor 32, the evaporative condenser 33, the ice storage tank 34, the water chiller 35, the high pressure receiver 69, the low pressure receiver 71 and the heat recovery unit 96 of the Fig. 15 apparatus have all been omitted from that of Fig. 17, while absorption

refrigeration apparatus indicated generally at 376 has been added. Exhaust gases from the engine-generator 316 are either vented from a stack 377 or circulated through the absorption apparatus 376 to furnish energizing heat, depending on the positions of dampers 378 and 379. On summer cycle heat from the absorption apparatus 376 is rejected in the cooling tower 272, being transferred thereto by water circulated to the apparatus 376 through lines 380 and 381, while chilled water from the apparatus 376 is delivered to the line 276, used as previously described, and returned to the apparatus 376 through the line 282.

When the ambient conditions are such that the cooling tower 272 is capable of providing water at a temperature of 64°F. (18°C.) or lower, the dehumidifier 264 of Figs. 15 and 17 is capable of producing air having a dry bulb temperature of about 90°F. (32°C.) and a moisture content of 31 grains of water vapor per pound of dry air; as a consequence, it is not then necessary for air discharged from the dehumidifier 264 to be conditioned in the dehumidifier 265. To take advantage of this situation, the apparatus includes a duct 382 which connects the duct 270 and the duct 271, by-passing the dehumidifier 265. Dampers 383 and 384 in the ducts 382 and 270 can be set to direct all or any part of the air leaving the dehumidifier 264 into the duct 382. Since the heat associated with dehumidification is transferred to cooling tower water from the dehumidifier 264 and is transferred to chilled water from the dehumidifier 265, it is usually economically advantageous to use the dehumidifier 264 to perform as much dehumidification as possible, and to minimize the use of the dehumidifier 265.

Sensible cooling of the dehumidified air of the apparatus of Figs. 15 and 17 is also possible, and is frequently advantageous. This can be done by a cooling coil (not illustrated) positioned in heat transfer relationship with air in at least one of the ducts 269, 270, 271, and 382. Heat can be transferred from the cooling coil to water flowing in the lines 275 and 273 to and from the cooling tower 272 or to chilled water flowing in the lines 278 and

280. Relatively high temperature dehumidified air, however, is desirable, as discussed above, because insulated ducts are not required to prevent condensation. Accordingly, it is usually preferred that the temperature of the air in the duct 271 be from about 58°F. (14°C.) to about 90°F. (32°C.).

It will be appreciated that the apparatus of Figs. 15-17 has an advantage which the apparatus of Figs. 1, 4, 5 and 14 lacks, namely, that the primary air is dehumidified, but relatively warm, so that its distribution does not necessitate the use of insulated ducts. Conversely, the apparatus of Figs. 1, 4, 5 and 14 has an advantage which the apparatus of Figs. 15-17 lacks; that advantage is minimum first cost in the case of the apparatus of Fig. 5, and, in the case of the apparatus of Figs. 1, 4 and 14, the ability to produce ice at night with low cost energy and to use the ice during the day to carry a substantial portion of the air conditioning load. An air handler indicated generally at 385 in Fig. 18 enables the use of ice or direct expansion refrigeration apparatus to produce extremely dry air at a sufficiently high temperature that it can be distributed in uninsulated ducts.

Return air from apparatus served by the air handler 385 flows through a return duct 386 and a return blower 387, while ambient air flows through a duct 388 and a louver 389 into the air handler 385. Some of the air from the blower 387 can be vented as relief air, leaving through an outlet 390, while the rest flows through vanes 391 and is mixed with ambient air. The mixture flows through the air handler in heat exchange relationship with coils 392 and 393, through a supply blower 394, in heat exchange relationship with a coil 395, and then exits in a duct 396. A pump 397 causes a heat transfer fluid to flow through a line 398, the coil 392, a line 399 and the coil 395 and back to the pump 397. The coil 393 is cooled to a low temperature, say 36° F. (2°C.); as a consequence, heat transfer fluid circulated by the pump 397 is cooled in the coil 395, transferring heat to conditioned air which enters the duct 396, and is warmed in

the coil 392, heat being transferred thereto from the mixture of return air and ambient air. The air in the duct 396 should be at a temperature of at least about 53°F. (14°C.) so that insulation is not required on risers, 5 headers, ducts and the like in which it is distributed. A safe temperature can be achieved by sizing the apparatus so that the temperature of the air in the duct 396 is about half way between the temperature, say 86°F. (30°C.), of the mixture of ambient air and return air, and the temperature, 10 say 45°F. (7°C.), of the air leaving the blower 394. The coil 393 is served by lines 400 and 401 through which a heat transfer fluid that has been cooled by heat transfer to stored ice as in the apparatus of Figs. 1 and 4 is circulated, or through which a refrigerant can be so 15 circulated, in which case the coil 393 is a direct expansion coil as in the Fig. 5 apparatus.

Three way valves 402 and 403 in the lines 398 and 399 can be used to divert coolant so that, instead of flowing through the coil 392, it flows through a line 404 to other 20 apparatus (not illustrated in Fig. 18), returning, after heat transfer thereto in the other apparatus, through a line 405 and the valve 403 to the line 399 and the coil 395. For example, the lines 404 and 405 can be connected so that heat is transferred to the heat transfer fluid circulated 25 therethrough from the water which flows through the coils 48 of the apparatus of Figs. 1, 4 and 14.

On winter cycle, heated fluid in the lines 372 and 375 is not required in the heat exchanger 625 of the apparatus of Figs. 15-17, because there is no need to regenerate 30 desiccant. Accordingly, a valve 406 can be closed, and the heat transfer fluid can flow through lines 407 and 408 to serve the heat exchanger 326 and through lines 409 and 410 to serve the heater 327. Heated fluid from the heat exchanger 326 flows through lines 411 and 412 to and from 35 the lines 281 and 277 to serve the heat exchange coils 302, the unitary heat pumps 348, or both, in the mixing boxes 266, the flow being as previously described.

It will be observed that there is a valve 413 in the line 281. To prevent any possibility of condensation on the sprinkler mains 308 and 309, on the conduits 303 and 307, on the coils 302, and the like, the valve 413 can be kept closed except when a signal from a humidistat (not illustrated in Figs. 15-17) indicates that the humidity is sufficiently low that there is no chance of condensation. A single humidistat can be used for an entire building, or for each humidity zone of the building, as previously discussed.

The apparatus of Figs. 15 and 17 can be modified by elimination of the chemical dehumidifiers 264 and 265, and of the duct 271, and substitution therefor of the direct expansion compression refrigeration apparatus of Fig. 5 which includes the compressor 142, and the associated equipment, including the line 151, the evaporative condenser 33, the line 152, the coil 40, the air handler 30, the line 153 and the duct 39. The modified apparatus supplies cold dehumidified air to the duct 39.

As has been indicated above, in the apparatus of Figs. 15-17, return air from the duct 334 enters the blower 335 from which regenerating air is discharged into the regenerator 290. While only one space 296 is shown, it will be appreciated that the return air in the duct 334 is from all of the spaces served by the air conditioning apparatus. Similarly, only a part of the return air ordinarily flows through the regenerator 290 as relief air, a part being used as recirculation air. In the apparatus of Figs. 15 and 17 the recirculation air flows through a duct 414 into the duct 269 on the suction side of the blower 267. In the apparatus of Fig. 16, the recirculation air flows through the duct 414 into the conduit 365 on the suction side of the blower 357. The rate of flow of recirculation air through the duct 414 is controlled by a damper 415. Typically, a mixture of partially dehumidified outside air and recirculation air may enter the blower 267 at rates, respectively, of up to 0.13 and up to 0.12 cubic foot per minute per square foot of floor space. Relief air at a rate of up to 0.13 cubic foot per minute per square foot of floor space (the same rate at

which partially dehumidified outside air enters the blower 267) is discharged through the regenerator 290 or, in the case of the apparatus of Fig. 16, through the regenerator 290, and through the conduit 368.

5 The apparatus of Fig. 19 includes most of the elements of the apparatus of Fig. 1, as is indicated by the use of the same reference numerals, including the air handler 30, and the refrigeration apparatus which comprises the compressor 32, the evaporative condenser 33 and the
10 evaporator which serves the ice storage tank 34; this evaporator operates to produce ice, usually on night cycle when the building served by the apparatus is unoccupied, while a second evaporator, as subsequently explained in detail, operates on day cycle at times when the electric
15 utility does not impose a demand charge.

Outside air can be directed through or by-passed around the indirect evaporative cooler 36, as indicated by the arrows 37 and 38, before it is conditioned in the air handler 30 and distributed through risers (not illustrated)
20 and ducts 39 (one of which is shown in Fig. 19) to the building. In the air handler 30, in one mode of operation, air is conditioned by contact with the coil 40 to a dry bulb temperature of substantially 42°F. (6°C.). Ice water from the ice storage tank 34 at, say, 38°F. (3°C.) is circulated
25 by the pumps 41, flowing through the line 42, the pumps 41, the line 43, the coil 40 and the line 44 back to the tank 34. The flow of ice water through the coil 40 is modulated to maintain the 42°F. (6°C.) temperature of the conditioned air leaving the air handler 30. Whenever the ambient air has
30 a low moisture content, it is economically desirable to use the indirect evaporative cooler 36 and, thereby, to reduce the requirement for ice water in the coil 40.

Conditioned air from the ducts 39 is delivered to perimeter mixing boxes 416 and interior mixing boxes 417 at
35 a rate which is caused to vary as required by the air conditioning load in the spaces served by the mixing boxes 416 and 417. The mixing boxes 416 are of the "fan/coil" type, having constant speed fans 418 and coils 419; they are

also of the unitary heat pump type, having coils 420 to which heat can be pumped from condensers 421 of first heat pumps and coils 422 from which heat can be pumped to evaporators 423 of second heat pumps. Lines 424 connect the
5 condensers 421 and the evaporators 423 to the lines 50 and 60. The mixing boxes 417 are of the induction type, having a plurality of induction nozzles 425, one of which is shown in Fig. 19, through which conditioned air from the ducts 39 flows, inducing a flow of recirculated air from the space or
10 from a plenum, as indicated by an arrow, through induced air inlets 426. The recirculated air mixes with the conditioned air in mixing portions 427 of the mixing boxes 417, so that it is a mixture of conditioned air from the ducts 39 and recirculated air that is delivered to the space from
15 discharge ends 428 of the mixing boxes 417.

The fans 418 of the mixing boxes 416 have a capacity greater than the maximum flow of conditioned air to the mixing boxes 416; as a consequence, air is caused to flow from a space served thereby into each of the mixing boxes,
20 where it is mixed with conditioned air. The mixture of air from the space and conditioned air is returned to the space from the fan discharge. The spaces served by the mixing boxes 416 are below, while the boxes 416 are above, ceilings 429. The air flow described above is indicated in Fig. 19 by
25 arrows 430 and 431, the latter representing the flow of a mixture of conditioned air and recirculated air from one of the mixing boxes 416 and the former representing the flow of air from the space into the mixing box 416.

Either chilled heat transfer fluid or evaporatively
30 cooled heat transfer fluid is delivered to the mixing boxes 416, being circulated by the pumps 52 through the line 54, the main header 55, the supply line 56, the header 57 of the first sprinkler grid, one of the several sprinkler conduits 58 of the first sprinkler grid, and the supply line 59, to
35 the mixing boxes 416 and returning through the return line 60, one of the several sprinkler conduits 61 of the second sprinkler grid, the header 62 of the second sprinkler grid, the return line 63, the main return 64 and the line 65 back

to the pumps 52. The heat transfer fluid circulated as just described is either chilled in the heat exchanger 90 by heat transfer therefrom to fluid flowing in the line 44 from the coil 40 to the ice storage tank 34 or is evaporatively cooled by heat transfer therefrom in the heat exchanger 85 to water that has been cooled in the evaporative cooler 82. When chilled water is delivered to the mixing boxes 416 it is circulated through the coils 419, and is at a comparatively high temperature, sufficiently high that moisture is not condensed when room air at design conditions flows over the coils 419. In a typical instance, the water in the coils 419 will be at 58°F. (14° C.), and the room air will be at 75°F. (24°C.) and 50% relative humidity. In this mode of operation, dampers 432 can be modulated as desired to control the flow of conditioned air from the ducts 39 into each of the mixing boxes 416, and valves 433 can be modulated by controllers 434 to maintain the temperature sensed by thermostats 435 within control limits. While the mixing boxes 416 are operating as just described, cooling will often be required in some perimeter zones of a building while heating is required in others. This can occur because of a solar load that is imposed on different perimeter zones at different times of the day, because of differences in occupancy, or because of differences in the use of lights or of heat generating electronic apparatus, to mention a few of the possibilities. The mixing boxes 416 are well suited to handle this situation because heat pumps associated with the condensers 421 can be energized where heat is required, and the valves 433 can be set so that the 58°F. (14°C.) water by-passes the associated coils 419; a heat transfer fluid is then circulated from the condensers 421 through lines 436, through the coils 420 and through lines 437 back to the condensers 421 so that heat is pumped from the circulated heat transfer fluid to the recirculated air where required. The mixing boxes 416 can be operated in the same way when ambient conditions are such that evaporatively cooled heat transfer fluid is available at 58°F. (14°C.).

It is sometimes desirable to circulate evaporatively cooled heat transfer fluid to the mixing boxes 416 even when ambient conditions are such that the temperature thereof is higher than 58°F. (14°C.). For example, if the building 5 served by the apparatus of Fig. 19 is occupied during a part of the time when the electric utility imposes no demand charge, it is less costly to use electricity during that time to carry the air conditioning load than it is to use stored ice. This can be done by circulating an evaporatively 10 cooled heat transfer fluid to the mixing boxes 416 and using the heat pumps associated with the condensers 421 or those associated with the evaporators 423 to pump heat to or from the recirculated air. Whenever the building is occupied at a time when the electric utility imposes no demand charge, it 15 is also more energy efficient, by comparison with the use of ice for the purpose, to circulate refrigerant from the low pressure receiver 71 through a line 438 to a DX coil 439 in the air handler 30 and through a line 440 back to the low pressure receiver 71; in this mode of operation, the 42°F. 20 (6°C.) air that is delivered through the duct 39 is produced by contact with the DX coil 439. During the time that there is a demand charge, then, ice produced during night cycle is used to provide a heat transfer fluid at 38°F. (3°C.) that is circulated from the ice storage tank 34 through the line 25 43 to the coil 40 in the air handler 30, returning through the line 44 and the heat exchanger 90 to the ice storage tank 34; heat is transferred in the exchanger 90 from fluid circulated by the pumps 52 to maintain its temperature at substantially 58°F. (14°C.). The extra friction introduced 30 into the system by the DX coil 439 must be taken into account in determining whether or not there is a net saving in energy as a consequence of its use; it will often be preferable to save the energy necessary to overcome the friction rather than to save energy by using the DX coil.

35 A temperature sensor and controller 441 controls a damper 442 to vary the rate at which conditioned air from the ducts 39 enters each of the interior mixing boxes 417 as required to maintain a desired temperature within each of

the interior spaces, the minimum damper position being one which provides the minimum ventilation air. As long as the rate of flow of conditioned air into and through the mixing boxes 417 is sufficiently high, an adequate flow of 5 recirculated air is induced to flow without the need for a blower 443 to be energized. Whenever the flow of conditioned air is inadequate to cause the required induction in any of the mixing boxes 417, the blower 443 is energized to provide an adequate circulation of air at a temperature sufficiently 10 high that it does not cause discomfort. As is subsequently explained in more detail, air from the blowers 443 by-passes the induction nozzles 425 in the mixing boxes 417 but, in other functionally equivalent mixing boxes, blowers can discharge through induction nozzles; back draft dampers (not 15 illustrated in Fig. 19) prevent the flow of air except as described.

The apparatus of Fig. 20 includes some of the elements of the apparatus of Fig. 1, as is indicated by the use of the same reference numerals, including the air handler 30, 20 and the refrigeration apparatus which comprises the compressor 32, the evaporative condenser 33 and the evaporator which serves the ice storage tank 34; this evaporator operates to produce ice on night cycle or whenever the electric utility does not impose a demand 25 charge.

Outside air can be directed through or by-passed around the indirect evaporative cooler 36, as indicated by the arrows 37 and 38, before it is conditioned in the air handler 30 and distributed through risers (not illustrated) 30 and ducts (one of which is shown in Fig. 20, designated 39) to the building. In the air handler 30, air is conditioned by contact with the coil 40 to a dry bulb temperature of substantially 42°F. (6°C.), as described with reference to Fig. 1.

35 The Fig. 20 apparatus comprises a plurality of secondary air handlers 444, each of which is substantially identical with the previously described mixing boxes 31 of Figs. 1, 4, 5, 12 and 14-17, but is sized to serve a

plurality of zones of the building, often an entire floor. Each of the secondary air handlers 444 has a blower 445 which discharges air, as indicated by a head 446 of an arrow, which it induces to flow, as indicated by a tail 447 of an arrow, from an adjacent space. The discharge of air from the secondary air handlers 444 is into an associated duct 448, from which it is available through ducts 449 to each of a plurality of mixing boxes 450, which are of the dual duct type. Conditioned air from the ducts 39 is delivered through a duct 451 and ducts 452 to the mixing boxes 450. The proportions in which conditioned air from the ducts 39 and air from the ducts 449 enter the mixing boxes 450 are controlled, respectively, by dampers 453 in the ducts 452 and dampers 454 in the ducts 453. The dampers 453 and 454 serving each of the mixing boxes 450 work in opposition under the control of thermostat-controllers 455 so that a substantially constant volume, variable temperature flow of air is delivered through air inlets 456 to the building zone served by each.

The apparatus of Fig. 20, operated as described in the preceding paragraph, requires more dehumidified air for temperature control than is necessary for humidity control, so that there is an operating cost penalty by comparison with the same apparatus operated so that only the amount of dehumidified air required for humidity control is used. However, there are both first cost and operating cost advantages by comparison with the conventional system described above where air cooled to about 55°F. (13°C.) is distributed as required for temperature control.

There are coils 457 in the secondary air handlers 444. On winter cycle, a warm heat transfer fluid can be circulated through the coils 457 so that warm air is available to the mixing boxes 450 as required for heating. The warm heat transfer fluid can be circulated from a heat exchanger (not illustrated) served, as required, by a boiler (not illustrated) through the building sprinkler system to each of the coils 457 and back to the heat exchanger.

As has been stated above, the dampers 432 of the apparatus of Fig. 19 can be controlled in any suitable manner. A particularly desirable way to control these dampers is by means of controllers 458 which modulate the 5 dampers 432 to keep the humidity sensed by humidstats 459 within control limits; when the dampers are so controlled, the temperature in the space served by each of the mixing boxes 416 can be controlled as previously described, i.e., by modulating the flow of cooled water through the coils 10 419, by pumping heat to the coils 420 or by pumping heat from the coils 422, as required.

The air handler 30 of Fig. 20 includes a coil 460 to which a relatively high temperature, say 58°F. (14°C.), heat transfer fluid can be circulated through lines 461 and 462 15 from a heat exchanger 463. The heat exchanger 463 is served by heat transfer fluid which flows through lines 464 and 465 from the sprinkler grid. Use of the coil 460 is advantageous, other factors being equal, because the relatively high temperature coolant is less expensive, per 20 ton of refrigeration, than the low temperature coolant that is circulated through the coil 40.

Figs. 19 and 20 show two modifications of the apparatus of Fig. 1. Analogous modifications of the apparatus of Figs. 4, 5, 12 and 14-17 can also be made, and 25 the control device of Fig. 10 can be used in the apparatus of Figs. 19-35, as can the humidistat 200 where humidity conditions are suitable.

The apparatus of Figs. 5 and 12 can be modified by substituting other apparatus for the refrigeration apparatus 30 which includes the compressor 141. An example of apparatus where such substitution has been made is shown in Figs. 21 and 22. In the Fig. 21 apparatus, cooled or warmed water for circulation through the main header 55, the sprinkler grids, the cooling coils 48, the main return 64 and back to the 35 header 55 is provided by an absorption chiller/heater indicated generally at 466 and comprising a heater 467 and absorption refrigeration apparatus which includes an evaporator 468. Gas enters the absorption chiller/heater 466

as indicated by an arrow 469 and is burned, providing heat, which is either transferred to water delivered through a line 470 to the chiller/heater 466 from the line 64 and returned through a line 471 to the line 55 or used to energize absorption refrigeration apparatus which includes the evaporator 468 to which heat is transferred from water which is delivered thereto from the line 64 through a line 472 and returned through a line 473 to the main header 55. When the absorption refrigeration apparatus of the chiller/heater 466 is used, heat from the absorber and from the condenser thereof is transferred to a cooling tower 474.

In the Fig. 22 apparatus, water from a closed circuit evaporative cooler 475 is the sole means for removing heat from the water circulated through the line 55, the sprinkler grids and the line 64 back to the line 55. This water is supplied, however, to the cooling coils 419, to the condensers 421 or to the evaporators 423 of the mixing boxes 416, and heat is pumped to the coils 420 or from the coils 422 where heating or additional cooling of the recirculated air is required, all as previously discussed in connection with Fig. 19.

Apparatus otherwise similar to that of Fig. 1, but which includes the air handler 385 of Fig. 18 and mixing boxes 476 is shown in Fig. 23. The mixing boxes 476 are of the "fan/coil" type, having constant speed fans 477 and coils 478; they are also of the unitary heat pump type, having coils 479 to which heat can be pumped from condensers 480 of first heat pumps and coils 481 from which heat can be pumped to evaporators 482 of second heat pumps.

The fans 477 of the mixing boxes 476 have a capacity greater than the maximum flow of conditioned air to the mixing boxes 476; as a consequence, air is caused to flow from a space served thereby into each of the mixing boxes, where it is mixed with conditioned air. The mixture of air from the space and conditioned air is returned to the space from the fan discharge. The spaces served by the mixing boxes 476 are below, while the boxes 476 are above, ceilings 483. The air flow described above is indicated in Fig. 23 by

arrows 484 and 485, the latter representing the flow of a mixture of conditioned air and recirculated air from one of the mixing boxes 476 and the former representing the flow of air from the space into the mixing box 476.

5 Either chilled heat transfer fluid or evaporatively cooled heat transfer fluid is delivered to the mixing boxes 476, being circulated by the pumps 52 through the line 54, the main header 55, the supply line 56, the header 57 of the first sprinkler grid, one of the several sprinkler conduits 10 58 of the first sprinkler grid, the supply line 59, to the mixing boxes 476 and returning through the return line 60, one of the several sprinkler conduits 61 of the second sprinkler grid, the header 62 of the second sprinkler grid, the return line 63, the main return 64 and the line 65 back 15 to the pumps 52. The heat transfer fluid circulated as just described is either chilled in the heat exchanger 90 by heat transfer therefrom to fluid flowing in the line 44 from the coil 40 to the ice storage tank 34 or is cooled by heat transfer therefrom in the heat exchanger 85 to water that 20 has been cooled in the evaporative cooler 82. When chilled water is delivered to the mixing boxes 476 it is circulated through the coils 478, and is at a comparatively high temperature, sufficiently high that moisture is not condensed when room air at design conditions flows over the 25 coils 478. In a typical instance, the water in the coils 478 will be at 58°F. (14° C.), and the room air will be at 75°F. (24°C.) and 50% relative humidity. In this mode of operation, dampers 486 can be modulated as desired to control the flow of conditioned air from the ducts 39 into 30 each of the mixing boxes 476, and valves 487 can be modulated by controllers 488 to maintain the temperature sensed by thermostats 489 within control limits. While the mixing boxes 476 are operating as just described, cooling will often be required in some perimeter zones of a building 35 while heating is required in others. This can occur because of a solar load that is imposed on different perimeter zones at different times of the day, because of differences in occupancy, or because of differences in the use of lights or

of heat generating electronic apparatus, to mention a few of the possibilities. The mixing boxes 476 are well suited to handle this situation because heat pumps associated with the condensers 480 can be energized where heat is required, and the valves 487 can be set so that the 58°F. (14°C.) water by-passes the associated coils 478; a heat transfer fluid is then circulated from the condensers 480 through lines 490, through the coils 479 and through lines 491 back to the condensers 480 so that heat is pumped from the circulated heat transfer fluid to the recirculated air where required. The mixing boxes 476 can be operated in the same way when ambient conditions are such that evaporatively cooled heat transfer fluid is available at 58°F. (14°C.).

The mixing boxes 476 are also of the induction type, having a plurality of induction nozzles 492, one of which is shown in Fig. 23, through which conditioned air from the ducts 39 or a mixture of such air with air discharged by the fans 477 flows, inducing a flow of recirculated air from the space or from a plenum, as indicated by an arrow, through induced air inlets 493. It is advantageous for controllers 494 to modulate the dampers 486 to maintain the humidity sensed by humidistats 495 within control limits. The humidistats 495 are positioned in the induced air inlets 493 where they detect the humidity of air induced to flow from the spaces they serve. Since space air is induced to flow into the inlets 493 whether or not the fans 477 are energized, humidity control can be maintained whenever the apparatus is operating while the fans 477 are energized only when they are needed to provide task cooling or heating. For example, a motion sensor (not illustrated) can be used in conjunction with the mixing boxes 476; whenever there is no motion in the space served by a given one of the mixing boxes 476, the fan 477 and the heat pumps which serve the condensers 480 and the evaporators 482 therein can be de-energized and the valve 487 can be set so that there is no flow of water through the coil 478. The controller 494 continues to modulate the damper 486 to maintain a desired humidity even when the space served by a given one of the

mixing boxes 476 is not occupied. As a consequence, as soon as motion is sensed in a previously unoccupied space, the fan 477 in the mixing box 476 which serves that space can be energized and chilled water can be used as previously described in connection with Fig. 19 with respect to the operation of the mixing boxes 416 to provide task heating or cooling.

Some spaces in a building are frequently occupied when the air conditioning system which serves the building is not in operation. The apparatus of Fig. 23 is well suited to provide air conditioning for the spaces that are occupied at such times. The air handler 385 can be operated to dehumidify air which is circulated on demand for ventilation and humidity control of the spaces that are occupied, and a heat transfer fluid can be circulated between the coils 392 and 393 of the air handler 385 so that the dehumidified air is essentially neutral in temperature, say 70°F. (21°C.). The fans 477 can then be energized in the mixing boxes 476 which serve the occupied spaces, and heat can be pumped from the coils 481 of those boxes for temperature control, as required. The heat transfer fluid can be circulated through the sprinkler system in this mode of operation, acting as a heat sink for condenser heat.

It will be appreciated that, in some instances, it will not be necessary to operate both of the heat pumps which serve the coils 479 and 481 of the mixing boxes 476. For example, it is often possible to design the apparatus so that, when it is in cooling mode, modulating the flow of chilled water through the coils 778 will enable the mixing boxes 476 to maintain a desired temperature as heat gains in the spaces served vary from maximum to minimum. In other cases, modulating the flow of chilled water through the coils 478 and either pumping heat to the coils 479 or pumping heat from the coils 481 will enable the boxes 476 to maintain a desired temperature. Accordingly, one or both of the coils 479 and 481, and the associated heat pumps, can sometimes be omitted from the mixing boxes 476.

Apparatus otherwise similar to that of Fig. 21, but in which the compression refrigeration apparatus which includes the compressor 142 has been replaced by a chemical dehumidifier 496, compression refrigeration apparatus which includes a compressor 497 and associated equipment is shown in Fig. 24. Refrigerant flows from the compressor 497, which is driven by a gas engine 498, to an evaporative condenser 499, to a DX coil 500, a DX coil 501, and back to the compressor 497, the flow being through lines 502, 503, 504 and 505. The DX coils 500 and 501 are a part of the air handling portion of the apparatus, the former being in a first air handler 506 and the latter being in a second air handler 507. Return air enters the first air handler 506 from the duct 201, some being vented through an outlet 508, and the rest flowing through an inlet 509 and being mixed with outside air which has been either directed through or by-passed around the indirect evaporative cooler 36, as indicated by the arrows 37 and 38. The mixture of outside air and recirculated air then flows in heat exchange relationship with the DX coil 500, through a duct 510, through the dehumidifier 496, through a duct 511, through the second air handler 507, a duct 512, and a washer 513 and into the ducts 39. In the second air handler 507 the air is in heat exchange relationship first with a coil 514 and then with the DX coil 501. The air is dehumidified in the dehumidifier 496 by contact with a concentrated hygroscopic liquid, e. g., alumina, silica or paper impregnated with lithium chloride, and is cooled by heat exchange first with the coil 514 and then with the coil 501. Evaporatively cooled water from a cooler 515 is circulated through the coil 514. The dehumidifier 496 is a wheel which rotates as indicated by an arrow 516 so that the air being dehumidified passes through successive segments of the wheel as they are advanced by rotation while regenerating air passes, as subsequently described, through different successive segments as they are advanced.

As an example of the operation of the apparatus of Fig. 24, outside air having a dry bulb temperature of 95°F.

(35°C.) and containing 99 grains of water vapor per pound of dry air can be mixed with return air to produce a mixture having a dry bulb temperature of 90°F. (32°C.) and containing 90 grains of water vapor per pound of dry air. This mixture 5 can then be cooled to a dry bulb temperature of 51°F. (11°C.) by contact with the coil 500 and dehumidified to a moisture content of 51 grains of water vapor per pound of dry air. In the dehumidifier 496 the air can be dehumidified to a moisture content of 10 grains of water vapor per pound 10 of dry air and heated to a dry bulb temperature of 100°F. (38°C.). This air can then be cooled sensibly by contact with the coil 514 to a dry bulb temperature of 95°F. (35°C.) and by contact with the coil 501 to a dry bulb temperature of 57°F. (14°C.) without, in either case, affecting its 15 moisture content. Finally, the air can be washed adiabatically in the washer 513 so that it enters the ducts 39 at a dry bulb temperature of 40°F. (4°C.) and containing about 37 grains of water vapor per pound of dry air.

It will be appreciated from the foregoing example of 20 its operation that the apparatus of Fig. 24 can be used to produce the low temperature, dry air that can be circulated in a small quantity as described above to achieve substantial savings in the original construction cost of air conditioning apparatus according to the instant invention. 25 The Fig. 24 apparatus differs from that previously described because it accomplishes this result using gas as an energy source, and without requiring electricity from a utility or either ice or desiccant storage. As is indicated by an arrow 517, gas enters the engine 498 as a fuel; the gas is 30 converted by the engine 498 to shaft work which drives the compressor 497 and heat in the form of hot gases. The hot gases flow through a segment of a heat exchanger 518 and are vented while a blower 519 directs air through the other side of the heat exchanger 518 and through a segment of the 35 dehumidifier 496 to effect regeneration of that segment. Rotation of the dehumidifier causes successive segments thereof to present themselves for regeneration. It will be appreciated that a diesel or other combustion engine could

be used in place of the gas engine 498, and that a gas turbine, diesel or other engine could also be use to drive an electric generator to power an electric motor to drive the compressor 497. Where a combustion engine which has a cooling jacket is used, heat from the jacket is available in addition to heat from the combustion products.

Further, the gas turbine or other engine could be sized to provide the heat required by the absorption chiller heater 466.

Apparatus similar to that of Fig. 4, except that the mixing boxes 31 have been replaced by mixing boxes 519, is shown in Fig. 25. Conditioned air from the ducts 39 is delivered to the mixing boxes 519 at a rate which varies, depending upon the settings of individual dampers 520, each of which is actuated by a thermostat/humidistat-controller 521. The mixing boxes 519 are of the "fan/coil" type, having constant speed fans 522, coils 523 and coils 524. The fans 522 have a capacity greater than the maximum flow of conditioned air to the mixing boxes 519 when the dampers 520 are in their full open positions; as a consequence, there is a flow of recirculated air therethrough as previously described and as indicated by an arrow having the head 50 and the tail 51.

The thermostat/humidistat controllers 521 actuate the dampers 520 to establish and maintain a desired humidity in the space served by each of the mixing boxes 519, opening the dampers when the humidity is too high, and closing them when the humidity is too low. The minimum damper settings are those at which each of the mixing boxes 519 furnishes the minimum ventilation air. When humidity control has been established and the flow of conditioned air at the rate required to maintain the desired humidity is insufficient to counteract heat gains in the space served by one of the mixing boxes 519, a three-way valve 525 is set by the thermostat/humidistat controller 521 to cause chilled water circulated by the pumps 52 as previously described to flow through the coil 523 in that box, and the rate of flow is modulated by the valve 66 which is set as required by the

thermostat/humidistat controller 521 to maintain the desired temperature. When the flow of conditioned air at the rate required for humidity control is more than sufficient to counteract heat gains in the space served by one of the 5 mixing boxes 519, the thermostat/humidistat controller 521 sets the three-way valve 525 to cause chilled water to flow through the coil 524 in that box, and actuates the valve 66 in that box to modulate the flow of chilled water through the coil 524 to maintain the set temperature, increasing the 10 flow when the temperature is too low and vice versa. The chilled water is used to counteract heat gains when it is circulated through the coils 523, and for reheat when it is circulated through the coils 524. This is possible because the water is at about 58°F. (14°C.) while the room air which 15 flows in heat exchange relationship with the coils 523 is at about 75°F. (24°C.) and the conditioned air which flows in heat exchange relationship with the coils 524 is at about 40°F. (4°C.)

The mixing boxes 519 also include electric heaters 526 20 positioned for heat exchange with air from the space that is caused to flow therethrough. The heaters 526 can be used in place of or to supplement the coils 524 when reheat is required. Similarly, the mixing boxes 519 can include electric heaters (not illustrated) positioned for heat 25 exchange with conditioned air or with a mixture of conditioned air and recirculated air, and any of the heaters, or any combination of the heaters, can be used in place of or to supplement the coils 524 for reheat. It is also possible to circulate warm water to the coils 523 of 30 the mixing boxes 519 or to the coils 48 of the mixing boxes 31 (see, for example, Fig. 1) as required for reheat, but this requires a second circulating system and, therefore, usually is economically undesirable.

Apparatus which is the same as that of Fig. 25 except 35 that the mixing boxes 519 have been replaced by mixing boxes 527 is shown in Fig. 26. The mixing boxes 527 are of the "fan/coil" type, having the fans 522 and the coils 524, but they are controlled by thermostat controllers 528 which

modulate the dampers 520 for temperature control between settings that provide the minimum ventilation air and full open positions; whenever the setting that provides the minimum ventilation air more than counteracts heat gains in a given space, the controller 523 for the mixing box 527 which serves that space modulates the valve 66 of that box as required to provide the requisite reheat.

Apparatus which is the same as that of Fig. 25 except that the mixing boxes 519 have been replaced by mixing boxes 529 is shown in Fig. 27. The mixing boxes 529, which are controlled, as is subsequently explained in more detail, by thermost/humidistat-controllers 530, have heat pipes indicated generally at 531 and 532. The heat pipe 531 has a condensing section 533, an evaporating section 534, a vapor pipe 535, a liquid return line 536 and a pump 537 in the liquid return line 542. The pump 537 is operable to pump condensate from the condensing section 533 to the evaporating section 534. A valve 538 controls the operation of the heat pipe 531. The heat pipe 532 has a condensing section 539, an evaporating section 540, a vapor pipe 541, a liquid return line 542 and a pump 543 in the liquid return line 542. The pump 543 is operable to pump condensate from the condensing section 539 to the evaporating section 540. A valve 544 controls the operation of the heat pipe 532.

When the mixing boxes 529 are operating, the dampers 520 are modulated by the thermostat/humidistat-controllers 530 as required for humidity control. When cold primary air at the rate of flow required to control humidity is insufficient to counteract heat gains in the space served by one of the mixing boxes 529, the relevant thermostat/humidistat-controller 530 senses a temperature above the set point and, in response, activates the associated heat pipe 531 by energizing the pump 537 and opening the valve 538 thereof. The liquid of the heat pipe 531 is then pumped into the evaporating section 533, where it is vaporized by heat transferred thereto from air flowing through the mixing box from the space. The vapor which results flows through the vapor pipe to the condensing

section 533 where it is condensed by heat transfer therefrom to air in the plenum with which it is in heat transfer relationship. It will be appreciated that the heat pipe 531 must be in a cooled plenum to be capable of transferring 5 heat from recirculated air as just described: as previously described, the sprinkler systems of Figs. 6 and 7 can be used to cool the plenum to enable the heat pump 531 to operate. When one of the heat pipes 531 is not energized, the associated thermostat/humidistat-controller 530, in 10 response to a sensed temperature below the set point, activates the relevant one of the heat pipes 532 by energizing the pump 543 and opening the valve 544. The liquid of the heat pipe 532 is then pumped into the evaporating section 540, where it is vaporized by heat 15 transferred thereto from air in the plenum. The vapor which results flows through the vapor pipe to the condensing section 539 where it is condensed by heat transfer therefrom cold primary air flowing in heat transfer relationship therewith. The heat pipe 532 is capable of operating either 20 in a cooled plenum or in a plenum that is heated to a temperature several degrees above the space temperature because it is transferring heat to cold primary air.

The apparatus of Fig. 28 is the same as that of Fig. 27 except that the mixing boxes 529 have been replaced by 25 mixing boxes 545 which have heat pipes indicated generally at 546. The heat pipes 546 have a condensing section 547, an evaporating section 548, a vapor pipe 549, a liquid return line 550 and a pump 551 in the liquid return line 550. The pump 551 is operable to pump condensate from the condensing 30 section 547 to the evaporating section 548. A valve 552 controls the operation of the heat pipe 546.

The air handlers 30 of the apparatus of Figs. 25-28 have coils 553 connected by lines 554 and 555 to the headers 57 and 62. Relatively high temperature water in the the 35 coils 553 can carry a substantial proportion of the air conditioning load at a lower cost per ton of refrigeration, by comparison with the cost when lower temperature water from the ice storage tank 34 is used, provided that any

electricity used to produce the high temperature water does not contribute to a demand charge, for example, because the absorption apparatus 134 is used to cool the water, because electricity from the engine generator 79 is used, or because 5 electricity from a utility is used at a time when its use does not contribute to a demand charge. The apparatus of Figs. 27 and 28 does not use high temperature water to remove heat from air circulated through the mixing boxes 529 and 545; as a consequence, the only use for high temperature 10 water in the apparatus of these Figures is in the coils 553.

One of the mixing boxes 417 of the apparatus of Fig. 19 is shown in more detail in Figs. 29 and 30. Conditioned air enters the mixing box 417 through an inlet 556 at a rate which depends upon the setting of the damper 442 and is 15 discharged through the nozzles 425. When the rate of flow of conditioned air through the nozzles 425 is sufficiently high, recirculated air is induced to flow at a substantial rate through the induced air inlet 426, mixing with the conditioned air in the mixing portion 427. In this mode of 20 operation, a back-draft damper 557 prevents a flow of air from the nozzles 425 to the right in Fig. 29 through an air inlet 558, while the flow of air through the induced air inlet 426 opens a back-draft damper 559. When the flow of conditioned air to the mixing box 417 is throttled to such 25 an extent that its flow through the nozzles 425 is not capable of inducing an adequate flow of recirculated air through the inlet 426, the blower 443 is energized, inducing air to flow through the inlet 558 to the suction side of the blower 443; this air is discharged into a passage 560 which 30 bypasses the nozzles 425, forces the back-draft damper 559 to move to a "closed" position, and mixes with the conditioned air in the mixing portion 427. Accordingly, whether or not the blower 443 is energized, it is a mixture of conditioned air and recirculated air that is discharged 35 from the mixing box 417 into the space it serves.

A mixing box that is functionally equivalent to the mixing box 417 is designated 561 in Figs. 31 and 32. The mixing box 561 has a conditioned air inlet 562, induction

nozzles 563, a mixing portion 564, an air inlet 565, a blower 566, a back-draft damper 567, a conditioned air damper 568 (Fig. 32) and an induced air inlet 569 (Fig. 31). Conditioned air enters the mixing box 561 through the inlet 5 562 at a rate which depends upon the setting of the damper 568 and is discharged through the nozzles 563. When the rate of flow of conditioned air through the nozzles 563 is sufficiently high, this flow induces recirculated air to flow at a substantial rate through the induced air inlet 10 569, mixing with the conditioned air in the mixing portion 564. In this mode of operation, the back-draft damper 567 prevents a flow of air from the nozzles 563 to the right in Fig. 31 through the air inlet 565. When the flow of conditioned air to the mixing box 561 is throttled to such 15 an extent that its flow through the nozzles 563 is not capable of inducing an adequate flow of recirculated air through the inlet 569, the blower 566 is energized, inducing air to flow through the inlet 565 to the suction side of the blower 566; this air is discharged into a chamber 570 from 20 which it flows through the nozzles 563, forcing the back-draft damper 567 to move to an "open" position, and mixes with the conditioned air in the chamber 570 and with air induced to flow through the inlet 569 in the mixing portion 564. Accordingly, whether or not the blower 566 is 25 energized, air is induced to flow through the inlet 569 and it is a mixture of conditioned air and recirculated air that is discharged from the mixing box 561 into the space it serves.

Apparatus similar to that of Fig. 21, but wherein the 30 mixing boxes 39 have been replaced by mixing boxes 570, is shown in Fig. 33. The mixing boxes 570 are of the "fan/coil" type, having constant speed fans 571 and coils 572; they are also of the unitary heat pump type, having coils 573 to which heat can be pumped from condensers 574 of first heat 35 pumps and coils 575 from which heat can be pumped to evaporators 576 of second heat pumps; finally, they are of the induction type, having a plurality of induction nozzles 577, one of which is shown in Fig. 33, through which

conditioned air from the ducts 39 flows, inducing a flow of recirculated air from the space or from a plenum, as indicated by an arrow, through induced air inlets 578. Air which enters the mixing boxes 570 through the induced air inlets 578 mixes with air discharged from the induction nozzles 577 in mixing portions 579 of the mixing boxes 570, so that it is a mixture of these streams that is delivered to the spaces from discharge ends 580 of the mixing boxes 570.

10 The fans 571 of the mixing boxes 570 have a capacity greater than the maximum flow of conditioned air to the mixing boxes 570; as a consequence, when the fans 571 are operating, air is caused to flow through an air inlet 581 from a space served thereby into each of the mixing boxes, 15 where it is mixed with conditioned air. The mixture of air from the space and conditioned air flows through the induction nozzles 577, inducing a further flow of recirculated air through the induced air inlets 578; the air delivered to the spaces is a mixture of the air which flows 20 through the nozzles 577 and the the air that its flow induces. An arrow 582 indicates the flow of air through the air inlets 581, while an arrow 583 indicates the flow of an air mixture from the mixing boxes 570 to the spaces they serve.

25 Evaporatively cooled heat transfer fluid is delivered to the mixing boxes 570, being circulated thereto as previously described from the closed circuit evaporative cooler 475. This water is supplied to the cooling coils 572, to the condensers 574 or the evaporators 576, as required, 30 so that the required cooling can be done by the coils 572 or by the coils 575 or the required heating can be done by the coils 573. The apparatus also includes a coil 584 positioned for heat transfer with conditioned air before it flows through the nozzles 577. Heat transfer from this coil will 35 often provide all the reheat that is necessary, in which case the coils 573, the condensers 574 and the first heat pumps can be omitted. Similarly, chilled water can be circulated to the coils 572 and used as previously

described, and will often provide all of the supplemental cooling that is required, beyond that done by the conditioned air from the ducts 39.

The mixing box 570 is admirably suited for task cooling 5 when a damper 585 is controlled by a humidistat-controller 586 to maintain the humidity in a space it serves at a predetermined level while the operation of the fan 571, of the coils 572 and 584 and of the first and second heat pumps if they are present is controlled by a thermostat-controller 10 587 in cooperation with a signal indicating that the space served is occupied. The signal can be from a motion sensor (not illustrated) or can be one which an occupant of the space served actuates, e.g., by turning on the lights or by turning a separate switch to the on position. When 15 there is no signal indicating that the space is occupied, the fan 571 is not energized and the first and second heat pumps, if they are present, are not energized; as a consequence, the coils 572, 573 and 575 are essentially ineffective to counteract heat gains or losses in the space. 20 The coil 584, however, is operated by the controller 587 as previously described for reheat if the space temperature is below the set point. Whenever there is a signal which indicates that the space served is occupied, the fan 571 is operated and chilled or evaporatively cooled water is made 25 available to the coil 572 and to the condenser 574 and the evaporator 576 if the first and second heat pumps are used.

The mixing box 417 (Figs. 29 and 30) has a coil 588 and the mixing box 561 (Figs. 31 and 32) has a coil 589; either of these mixing boxes can be substituted for the mixing box 30 570 (Fig. 33) and operated as just described in the preceding paragraph when its coil (588 or 589) is connected between the lines 357 and 368, and chilled 58°F. (14°C.) heat transfer fluid is supplied to the sprinkler system as previously described.

35 Apparatus which includes many of the elements of that of Fig. 1 (but not the water chiller 35 and associated apparatus), and which additionally includes a secondary air handler 590 and mixing boxes 591 is shown in Fig. 34. The

secondary air handler 590 has an inlet 592 for recirculated air and a blower 593 which induces air from the zone served by the handler 590 to flow through the inlet 592 and discharges that air into a duct 594 from which it is delivered to the mixing boxes 591, flowing through ducts 595 at a rate which is determined by the settings of dampers 596. Conditioned air from one of the ducts 39 is also delivered to the mixing boxes 591, flowing thereto through ducts 597 at rates which depend upon the settings of dampers 598.

The mixing boxes 591 have coils 599 positioned for heat exchange with cold primary air entering from the ducts 597 and coils 600 positioned for heat exchange with recirculated air from the ducts 595. The flow of heat transfer fluid to the coils 599 and 600 is determined by the positions of valves 601 and of valves 602, respectively.

The dampers 596 and 598 and the valves 601 and 602 are controlled by humidistat controllers 603 and thermostat controllers 604. In operation, the dampers 598 are modulated as required to maintain a set humidity in the space served by each of the mixing boxes 591, and the dampers 596 are modulated in opposition to maintain a substantially constant flow of total air to the space served by each of the mixing boxes 591. When one of the thermostat controllers 604 senses a space temperature above the set point, it opens the associated one of the valves 602 to enable a heat transfer fluid at about 58°F. (14°C.) to flow through the associated coil 600, and modulates that valve as required to maintain the set temperature. Should the space temperature remain above the set point with the associated valve 600 in a full open position, the thermostat controller 604 overrides the associated humidistat controller 603 and modulates the dampers 596 and 598 in opposition to maintain the set temperature; during this time, the valve 602 is kept in its full open position. When one of the thermostat controllers senses a space temperature below the set point, it opens the associated one of the valves 601 and modulates that valve as required to maintain the set temperature. The valve 602 is

closed while the associated valve 601 is being modulated for reheat.

The apparatus of Fig. 34 is also admirably suited for task cooling. Whenever there is no signal indicating that the space served is occupied, the damper 596 serving that space is closed, and the associated damper 598 is modulated by the humidistat controller 603 as required for humidity control. If the thermostat controller 604 senses a temperature below the set point, it modulates the valve as required for reheat. As soon as there is a signal indicating that the space is occupied, operation as described above is resumed.

Apparatus which is the same as that of Fig. 25 except that the humidistat 200 and the controller 202 have been omitted and the thermostat controller 521 has been replaced by a humidistat/thermostat controller 605 is shown in Fig. 35. Each of the humidistat/thermostat controllers 605 controls the associated damper 45 as previously discussed to maintain the humidity of the space it serves within control limits, controls the coil 526 as required to maintain temperature when the amount of conditioned air required for humidity control is too little to overcome heat gains, and controls the coil 524 as required to maintain temperature when the amount of conditioned required for humidity control more than overcomes heat gains. Because the apparatus of Fig. 35 has no humidistat measuring the overall or average humidity of the building in which the mixing boxes 519 are situated, the option of using a single humidity reading to control the apparatus is not available.

Which apparatus is optimum for any given installation depends upon such factors as the local climate, including both temperatures and humidities and the local rate structures for electricity, gas and fuel oil, including not only cost per unit of energy, but also demand charges and incentives. In general, it is necessary to provide conditioned air at a sufficiently low humidity that only a small quantity thereof is required for humidity control, to deliver only a small quantity of the low humidity

conditioned air, and to circulate a heat transfer fluid, preferably, in most cases, through at least a part of a sprinkler system, for on site use, i.e., in or adjacent a space being conditioned, rather than in an equipment room, 5 to remove sensible heat. It is usually important to vary the rate at which the low humidity air is delivered so that humidity control is achieved, but over dehumidification is avoided. The low humidity conditioned air can be made by chemical dehumidification, using ice that was produced on 10 night cycle, or using a low temperature coil from which heat is transferred directly to the refrigerant of a refrigeration unit. Similarly, the heat can be removed from water that is circulated to carry the sensible heat load by absorption refrigeration apparatus, by compression 15 refrigeration, or with ice. When cogeneration is used, it is important to waste neither the shaft work nor the heat; the heat can be used on winter cycle for heating and on summer cycle either to regenerate a desiccant or as an energy source for absorption refrigeration apparatus, while the 20 shaft work can be used, summer and winter, either to generate electricity or to drive compressors, pumps, blowers and the like.

Most of the apparatus that is shown in the attached drawings transfers heat to evaporatively cooled water. This 25 is advantageous over transferring heat to water that has been chilled by refrigeration, because there are substantial savings in energy. However, ground water, for example from wells, when it is available, may also be at least equally advantageous, particularly in climates where high humidity 30 limits the use of evaporative cooling. When used, ground water should usually be circulated through a heat exchanger and returned to the ground. A suitably treated heat transfer fluid can then be chilled by heat exchange with the ground water and used in place of the evaporatively cooled water 35 that has been described above. For example, the apparatus of Fig. 17 can be modified by elimination of the dehumidifier 264 and of the cooling tower 272, and by connecting ground

water to the heat exchanger 312 and to the lines 380 and 381.

It is important that air conditioning apparatus introduce sufficient fresh or ventilation air into a building to prevent the accumulation of excessive concentrations of such inert gases as radon. Apparatus for determining the concentrations of such inert gases and for controlling ventilation air to keep their concentrations within safe limits is not presently available; occupants are not capable of detecting dangerously high concentrations of these gases. As a consequence, there is presently no mechanism for monitoring a variable to determine whether or not ventilation is adequate in a building. The apparatus of the instant invention makes the occupants of a building sensors to detect the inadequacy of ventilation; this occurs because the primary, conditioned air is relied upon to control humidity, and is circulated at a rate which is at least adequate for ventilation and at a sufficiently low moisture content that it also provides humidity control. If the apparatus is properly designed, and if it provides humidity control, it also provides adequate ventilation; if the apparatus fails to provide humidity control, ventilation may be inadequate, but the problem will be solved to quiet the complaints of the occupants.

It will be appreciated that various changes and modifications can be made from the specific details of the invention as shown in the attached drawings and described with reference thereto without departing from the spirit and scope thereof as defined in the appended claims.

For example, lithium chloride solutions have been described as aqueous desiccants, but other solutions are also operable, including other lithium halides, calcium chloride, and even glycol solutions. In one aspect, the invention involves the use of air conditioning apparatus to perform one function on day cycle and a different function on night cycle, one function during winter operation and a different function during summer operation, and minimizing the size of equipment required by storing what is made

during one mode of operation for use at a different time in a different mode of operation. For example, on summer operation, ice is produced on night cycle is used on day cycle to minimize energy requirements and to enable a given
5 air conditioning job to be performed with smaller equipment than would otherwise be required. Similarly, on winter-night cycle, heat is stored and ice is made; both are used on day cycle.

The apparatus of Fig. 21 can be modified by adding a
10 heat engine (not illustrated) to drive the compressor 32, and heat from the engine can be supplied to energize the absorption apparatus which includes the evaporator 468 or to heat the water which flows through the lines 470 and 471. Similarly, heat from the cogenerator 79 can be supplied to
15 energize the absorption apparatus which includes the evaporator 468 or to heat the water which flows through the lines 470 and 471. Also, the refrigeration apparatus which includes the compressor 32 can be replaced by a centrifugal package chiller which circulates a glycol, e.g., 30 to 50
20 percent by weight ethylene glycol.

It will be appreciated that apparatus which includes a reheat coil, e.g., the reheat coil 524 of Fig. 25, or a coil to which heat is pumped for reheat, e.g., the coil 420 of the mixing box 416 of Fig. 19 and a cooling coil should be
25 operated so that the cooling coil and the reheat coil do not operate at the same time as they would be, in essence, opposing one another. When cooling is required, reheat is not, and vice versa.

The apparatus of Fig. 18 can be modified by using a
30 heat pipe to transfer heat from air cooled by heat exchange with the coil 393 to incoming air. For example, the condensing section of a heat pipe can be substituted for the coil 395 and the evaporating section of the heat pipe can be substituted for the coil 392; a pump in a liquid return line
35 would then pump condensate from the condensing section to the evaporating section, and a valve in a vapor pipe would control the operation of the heat pipe.

For purposes of illustration some of the reheat coils, for example the coil 524 in Fig. 25, some of the dampers, for example the damper 486 in Fig. 23, and the condensing sections of some of the heat pipes, for example, the
5 condensing section 539 in Fig. 27, appear to be in ducts which serve associated mixing boxes. Ordinarily, the dampers, reheat coils and condensing sections would all be a part of the mixing boxes they serve, although it would also be possible for them to be contained in associated ducts.

10 The various cogenerators to which reference is made herein can be diesel engines, Otto cycle, or gas turbine (Brayton cycle) engines. A Stirling engine can also be used, with its shaft coupled directly to an electric generator or to a second Stirling engine, which then acts as a heat pump.

15

I Claim:

1. Air conditioning apparatus comprising a plurality of air outlets, means for dehumidifying air, means for
5 circulating dehumidified air to said air outlets, means operable to control its moisture content and temperature so that the dehumidified air is incapable, at the rate at which it is required for humidity control, of maintaining the desired space temperature at the maximum design cooling
10 load, a sensor for measuring the absolute or relative humidity of the space served by said air outlets, means responsive to said sensor, and operable to control the rate at which dehumidified air is delivered by each of said air outlets to the space it serves to maintain the moisture
15 content of the space served within control limits, and means operable to control the rate at which dehumidified air is delivered to said air outlets so that dehumidified air is available at the rate required from time to time by each of said air outlets.

20 2. Air conditioning apparatus as claimed in claim 1 wherein at least some of said air outlets are mixing boxes which additionally include means operable to cause a flow of recirculated air from the space, and, when dehumidified air is delivered thereto, to deliver to the space they serve a
25 mixture of dehumidified air and recirculated air.

3. Air conditioning apparatus as claimed in claim 2 which additionally includes means for cooling air recirculated by said mixing boxes, dehumidified air delivered to said mixing boxes, or the mixture of the two
30 before the mixture is delivered to the space by said mixing boxes.

4. Apparatus as claimed in claim 3 wherein said means for cooling air comprises a coil positioned for heat transfer with air recirculated by said mixing boxes, with
35 dehumidified air delivered to said mixing boxes, or with the mixture of the two, and which additionally includes cooling means for transferring heat from a heat transfer fluid circulated in heat exchange relationship therewith, and a

circulating system operatively associated with said coils and with said cooling means, for circulating a heat transfer fluid to and from said coils, and to and from said cooling means, said cooling means being operable to cool the heat transfer fluid to a temperature below the dry bulb temperature but above the dew point the apparatus is designed to maintain in the building.

5. Apparatus as claimed in claim 3 wherein said means for cooling air comprises a coil positioned for heat transfer with air recirculated by said mixing boxes, with dehumidified air delivered to said mixing boxes, or with the mixture of the two, and which additionally includes means for pumping heat from said coil to a heat sink.

6. Apparatus as claimed in claim 5 wherein said means for pumping heat from said coil to a heat sink is also operable to pump heat to said coil from a heat sink.

7. Air conditioning apparatus as claimed in claim 2 which additionally includes means for heating air recirculated by said mixing boxes, dehumidified air delivered to said mixing boxes, or the mixture of the two before the mixture is delivered to the space by said mixing boxes.

8. Apparatus as claimed in claim 7 wherein said means for heating air comprises a coil positioned for heat transfer with air recirculated by said mixing boxes, with dehumidified air delivered to said mixing boxes, or with the mixture of the two, and which additionally includes heating means for transferring heat to a heat transfer fluid circulated in heat exchange relationship therewith, and a circulating system operatively associated with said coils and with said heating means, for circulating a heat transfer fluid to and from said coils, and to and from said heating means.

9. Apparatus as claimed in claim 7 wherein said means for heating air comprises a coil positioned for heat transfer with air recirculated by said mixing boxes, with dehumidified air delivered to said mixing boxes, or with the

mixture of the two, and which additionally includes means for pumping heat to said coil from a heat sink.

10. Apparatus as claimed in claim 1 wherein said means for dehumidifying air comprises a coil, means for causing
5 air to be dehumidified to flow in heat exchange relationship with said coil, and means for transferring heat from said coil to dehumidify the air flowing in heat exchange relationship with said coil.

11. Apparatus as claimed in claim 10 wherein said means
10 for transferring heat from said coil comprises means for circulating a liquid having a low freezing temperature through said coil to remove heat from and to dehumidify the air flowing in heat exchange relationship with said coil.

12. Apparatus as claimed in claim 11 which additionally
15 includes means operable to cause a liquid which is miscible with water and forms therewith a liquid having a low freezing point to flow over said coil to prevent the accumulation of frost thereon.

13. Apparatus as claimed in claim 10 wherein said means
20 for transferring heat from said coil comprises direct-expansion compression refrigeration apparatus operatively connected to transfer heat from said coil to refrigerant of said refrigeration apparatus to cool and dehumidify the air which flows in heat exchange relationship with said coil.

25 14. Apparatus as claimed in claim 1 wherein said means for dehumidifying air includes a dehumidifier which employs a desiccant, and means for causing air to be dehumidified to flow in dehumidifying relationship with said dehumidifier.

15. Apparatus as claimed in claim 1 wherein said means
30 for dehumidifying air includes a dehumidifier which employs a desiccant, and which additionally includes a precooling coil, a postcooling coil, means for transferring heat from said precooling and postcooling coils, and means for causing air to be dehumidified to flow in heat exchange relationship
35 with said precooling coil, in dehumidifying relationship with said dehumidifier, and then in heat exchange relationship with said postcooling coil.

16. Apparatus as claimed in claim 15 wherein said means for transferring heat from said precooling and postcooling coils comprises compression refrigeration apparatus, and which additionally includes a combustion engine, means
5 operatively connecting said combustion engine in driving relationship with the compressor of said compression refrigeration apparatus, and means for transferring heat from said combustion engine to desiccant of said dehumidifier to enable regeneration thereof.

10 17. Apparatus as claimed in claim 16 wherein said combustion engine is a gas turbine.

18. Apparatus as claimed in claim 16 wherein said combustion engine is a diesel.

19. Apparatus as claimed in claim 1 wherein said means
15 for dehumidifying air includes a dehumidifier which employs a desiccant, and which additionally includes a precooling coil, a postcooling coil, and a washer, means for transferring heat from said precooling and postcooling coils, and means for causing air to be dehumidified to flow
20 in heat exchange relationship with said precooling coil, in dehumidifying relationship with said dehumidifier, in heat exchange relationship with said postcooling coil, and then through said washer.

20. Apparatus as claimed in claim 19 wherein said means
25 for transferring heat from said precooling and postcooling coils comprises compression refrigeration apparatus, and which additionally includes a combustion engine, means operatively connecting said combustion engine in driving relationship with the compressor of said compression
30 refrigeration apparatus, and means for transferring heat from said combustion engine to desiccant of said dehumidifier to enable regeneration thereof.

21. Apparatus as claimed in claim 20 wherein said combustion engine is a gas turbine.

35 22. Apparatus as claimed in claim 20 wherein said combustion engine is a diesel.

23. Apparatus as claimed in claim 1 wherein each of said air outlets is a mixing box which additionally includes

means operable to cause a flow of recirculated air from the space, and, when dehumidified air is delivered thereto, to deliver to the space it serves a mixture of dehumidified air and recirculated air, a cooling coil positioned for heat transfer with air recirculated by said mixing box, with 5 dehumidified air circulated to said mixing box, or with a mixture of the two before the mixture is delivered to the space by said mixing box, means for transferring heat from said cooling coil, and a thermostatic controller operably 10 connected to sense the temperature of the space served by said mixing box and to control the transfer of heat from said coil to maintain a control temperature in the space served by said mixing box.

24. Apparatus as claimed in claim 1 which additionally 15 includes an electric generator, a heat engine operably connected in driving relationship with said electric generator, and means operatively connecting said generator to introduce electricity into an electric grid which serves the building.

20 25. Apparatus as claimed in claim 2 wherein each of said mixing boxes is operably connected to deliver a mixture of dehumidified air and recirculated air to each of a plurality of second mixing boxes, wherein each of said second mixing boxes is operably connected to deliver the 25 mixture of dehumidified air and recirculated air to a space to be conditioned, and wherein each of said second mixing boxes includes means for varying the rate at which it delivers the mixture of dehumidified air and recirculated to the space it serves.

30

26. Apparatus as claimed in claim 10 wherein each of said mixing boxes is operably connected to deliver a mixture of dehumidified air and recirculated air to each of a plurality of second mixing boxes, wherein each of said 35 second mixing boxes is operably connected to receive dehumidified air and to deliver to the space it serves a mixture of the dehumidified air with the mixture of dehumidified air and recirculated air, and wherein each of

said second mixing boxes includes means for varying the rates at which it receives (1) the dehumidified air and (2) the mixture of dehumidified air and recirculated air.

27. An air outlet comprising an inlet for dehumidified
5 air, means for delivering dehumidified air supplied to said inlet to a space to be conditioned, a sensor for measuring the absolute or relative humidity of the space served by said air outlet, and means responsive to said sensor and to the pressure of dehumidified air supplied to said inlet, and
10 operable to control the rate at which dehumidified air is delivered by said air outlet to the space it serves to maintain the moisture content of the space served within control limits.

28. Air conditioning apparatus comprising a mixing box
15 to serve each of a plurality of spaces to be air conditioned, each of said mixing boxes having an outlet and a conditioned air inlet and being operable to deliver air through said outlet to condition the space it serves and to cause air introduced into said conditioned air inlet to flow
20 through through said outlet to the space, each of said mixing boxes also having a heat transfer device and a fan operable to induce air to flow from the space, in heat exchange relationship with said heat transfer device and then through the outlet to the space, said apparatus also
25 comprising means for dehumidifying air, means operable to control its moisture content and temperature so that the dehumidified air is incapable, at the rate at which it is required for humidity control, of maintaining the desired space temperature at the maximum design cooling load, means
30 for delivering dehumidified air to the conditioned air inlet of each of said mixing boxes, means including a humidistat operable to increase or decrease the rate at which dehumidified air is delivered to each of said mixing boxes as required to maintain a predetermined humidity, and means
35 including a thermostat operable to increase or decrease the rate at which heat is transferred by each of said heat transfer devices to or from air circulated in heat exchange

relation therewith to maintain a predetermined temperature in each of the spaces.

29. Apparatus as claimed in claim 28 which additionally includes means for withdrawing air from each of the spaces 5 and venting a part of the withdrawn air, and wherein said means including a humidistat is operable to sense the humidity of the withdrawn air and to increase or decrease the rate at which dehumidified air is delivered to each of said mixing boxes as required to maintain the withdrawn air 10 at a predetermined humidity.

30. Apparatus as claimed in claim 28 wherein said heat transfer devices are coils, wherein said apparatus includes means for chilling water and a system for circulating the chilled water to said coils, and which additionally includes 15 a control for said circulating system, said control being effective in a first position and ineffective in a second position to prevent the circulation of chilled water to said coils.

31. Apparatus as claimed in claim 30 wherein said 20 control is responsive to a signal from a humidistat and assumes the second position in response to such a signal indicating that the sensed humidity is below a predetermined level.

32. Apparatus as claimed in claim 31 which additionally 25 includes means for withdrawing air from each of the spaces and venting a part of the withdrawn air, and wherein said means including a humidistat is operable (1) to sense the humidity of the withdrawn air and to increase or decrease the rate at which dehumidified air is delivered to said 30 mixing boxes as required to maintain the withdrawn air at a predetermined humidity and (2) to transmit a signal from said humidistat to said control.

33. Apparatus as claimed in claim 28 which additionally includes a control for said fans, said control being 35 effective in a first position and ineffective in a second position to prevent the operation of said fans.

34. Apparatus as claimed in claim 33 wherein said control is responsive to a signal from a humidistat and

assumes the second position in response to such a signal indicating that the sensed humidity is below a predetermined level.

35. Apparatus as claimed in claim 34 which additionally includes means for withdrawing air from each of the spaces and venting a part of the withdrawn air, and wherein said means including a humidistat is operable (1) to sense the humidity of the withdrawn air and to increase or decrease the rate at which dehumidified air is delivered to each of said mixing boxes as required to maintain the withdrawn air at a predetermined humidity and (2) to transmit a signal from said humidistat to said control.

36. Apparatus as claimed in claim 28 wherein said heat transfer devices are coils, wherein said apparatus includes means for chilling water and a system for circulating the chilled water to said coils, and which additionally includes a control for each of said coils, each of said controls being effective in a first position and ineffective in a second position to prevent the circulation of chilled water to the associated one of said coils.

37. Apparatus as claimed in claim 36 wherein each of said controls is responsive to a signal from a motion sensor and assumes the second position in response to such a signal indicating that there is motion in the space served by the associated one of said coils.

38. Apparatus as claimed in claim 28 which additionally includes a control for each of said fans, said control being effective in a first position and ineffective in a second position to prevent the operation of the associated one of said fans.

39. Apparatus as claimed in claim 38 wherein each of said controls is responsive to a signal from a motion sensor and assumes the second position in response to such a signal indicating that there is motion in the space served by the associated one of said fans.

40. Apparatus as claimed in claim 28 wherein said heat transfer devices are coils, wherein said apparatus includes means for chilling water and a system for circulating the

chilled water to said coils, and which additionally includes a control for each of the mixing boxes, each of said controls being effective in a first position and ineffective in a second position to prevent the operation of the fan and the circulation of chilled water to the coil of the mixing box with which said control is associated.

41. Apparatus as claimed in claim 40 wherein each of said controls is responsive to a signal from a motion sensor and assumes the second position in response to such a signal indicating that there is motion in the space served by the associated one of said coils.

42. Apparatus as claimed in claim 28 wherein said mixing boxes additionally include induction means and are operable to cause dehumidified air which enters said conditioned air inlets to flow through said induction means and to induce a flow of air from the space into the mixing box, mixture of the induced air with the dehumidified air, and delivery of the mixture of induced air and dehumidified air to the space.

43. Apparatus as claimed in claim 15 wherein said means for transferring heat from said precooling and postcooling coils comprises compression refrigeration apparatus having a compressor driven by an electric motor, and which additionally includes a combustion engine operatively connected in driving relationship with an electric generator, and means operatively connecting said generator to introduce electricity into an electric grid which serves the building.

44. Apparatus as claimed in claim 19 wherein said means for transferring heat from said precooling and postcooling coils comprises compression refrigeration apparatus having a compressor driven by an electric motor, and which additionally includes a combustion engine operatively connected in driving relationship with an electric generator, and means operatively connecting said generator to introduce electricity into an electric grid which serves the building.

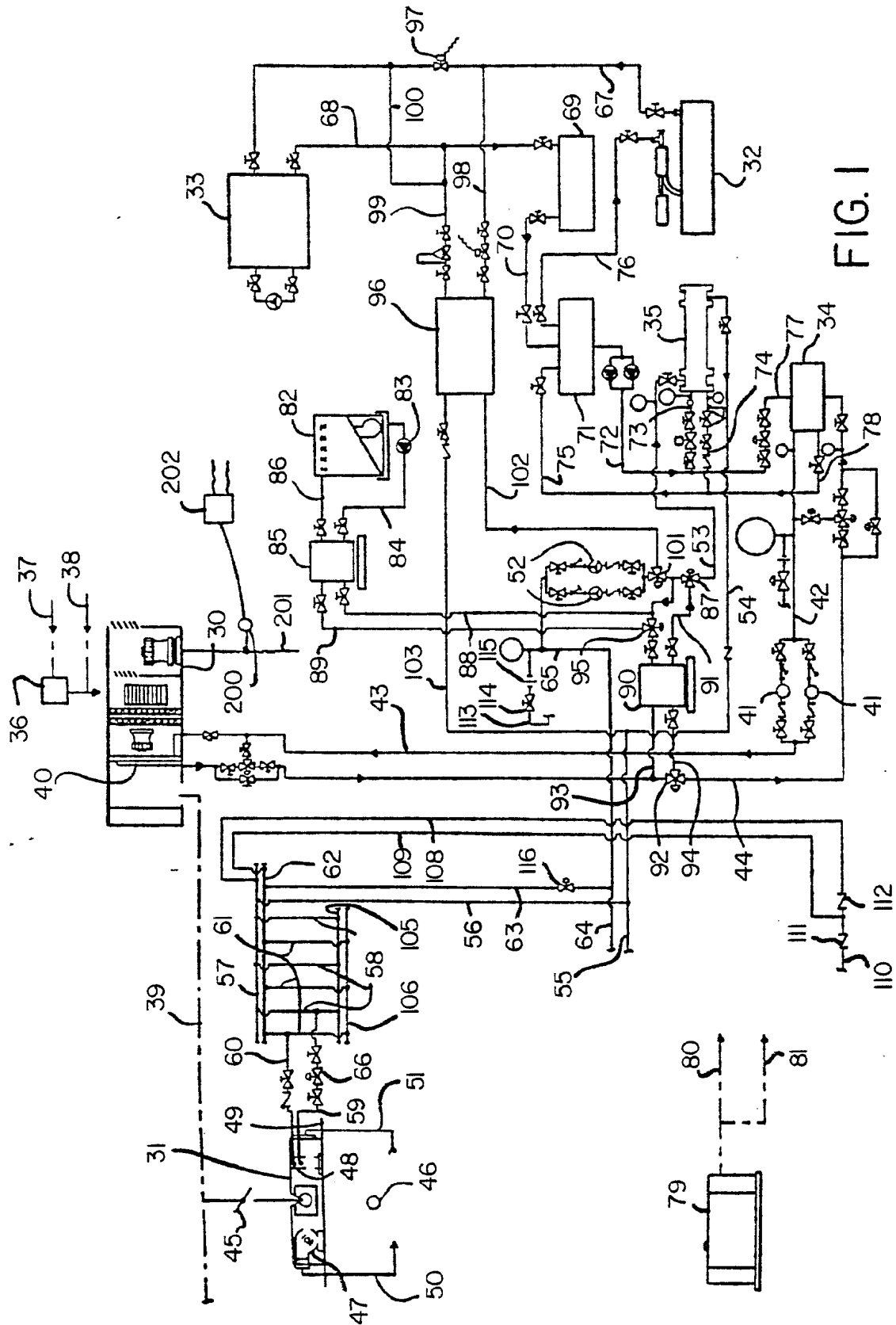


FIG. 1

FIG. 2

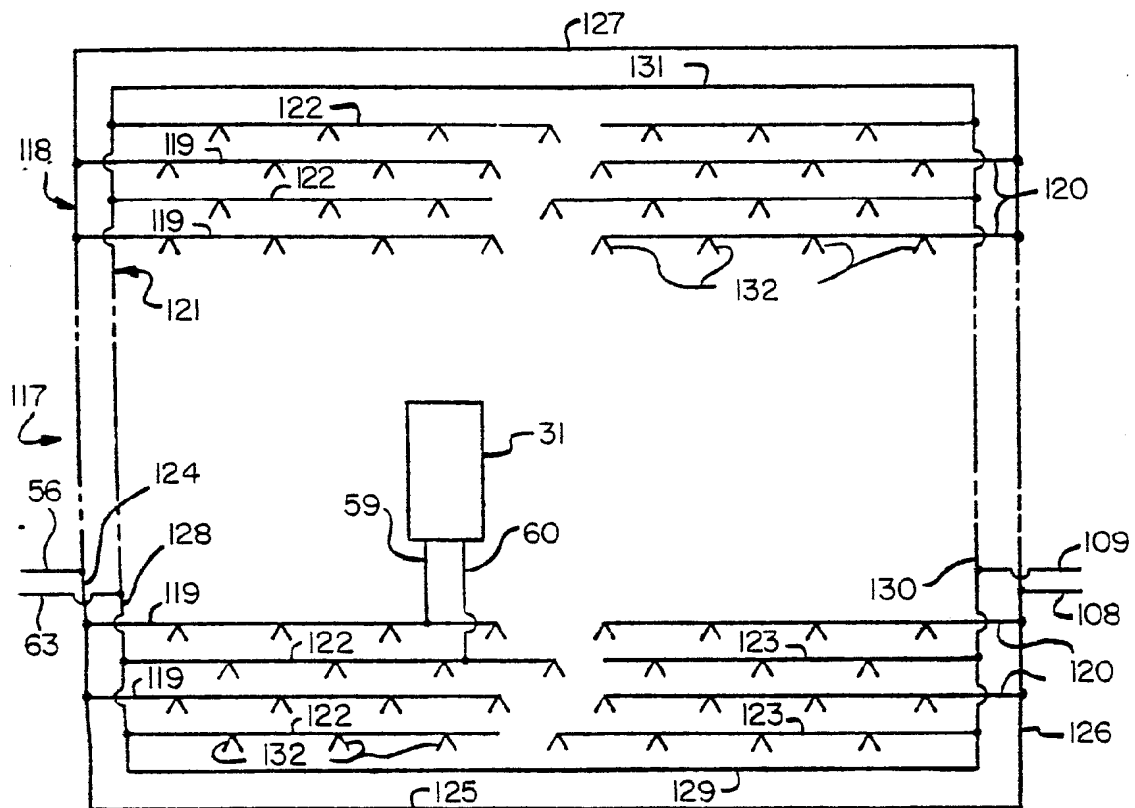
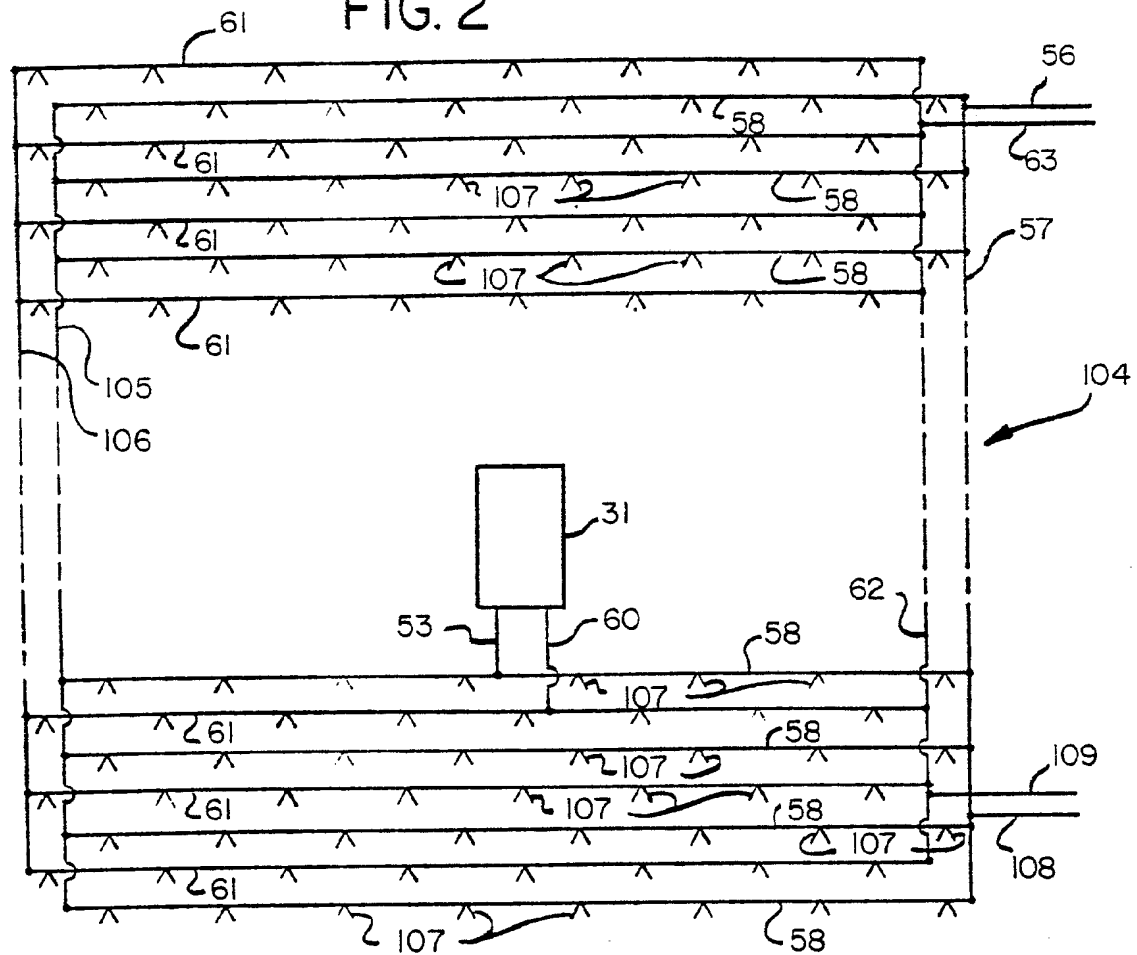


FIG. 3

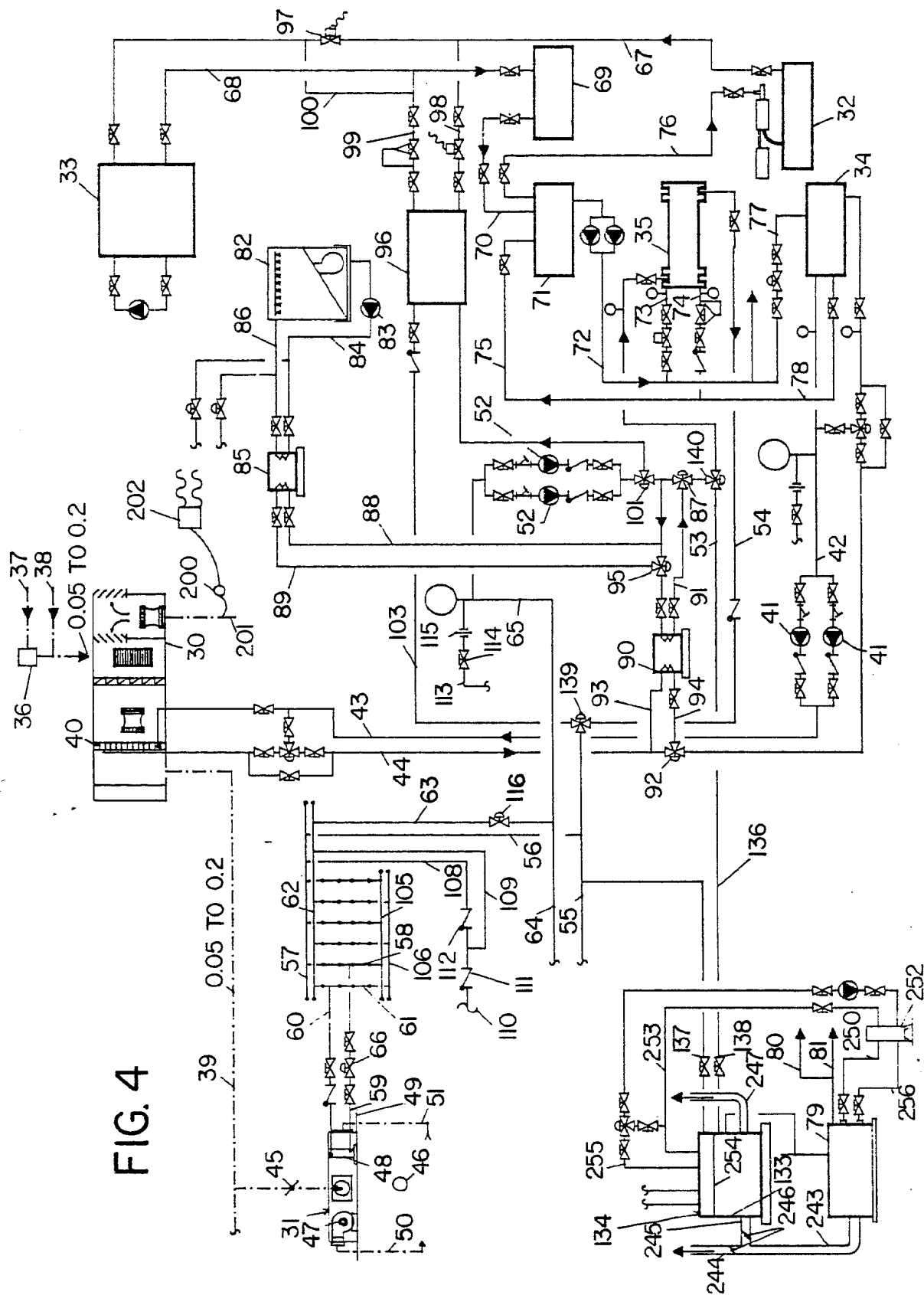
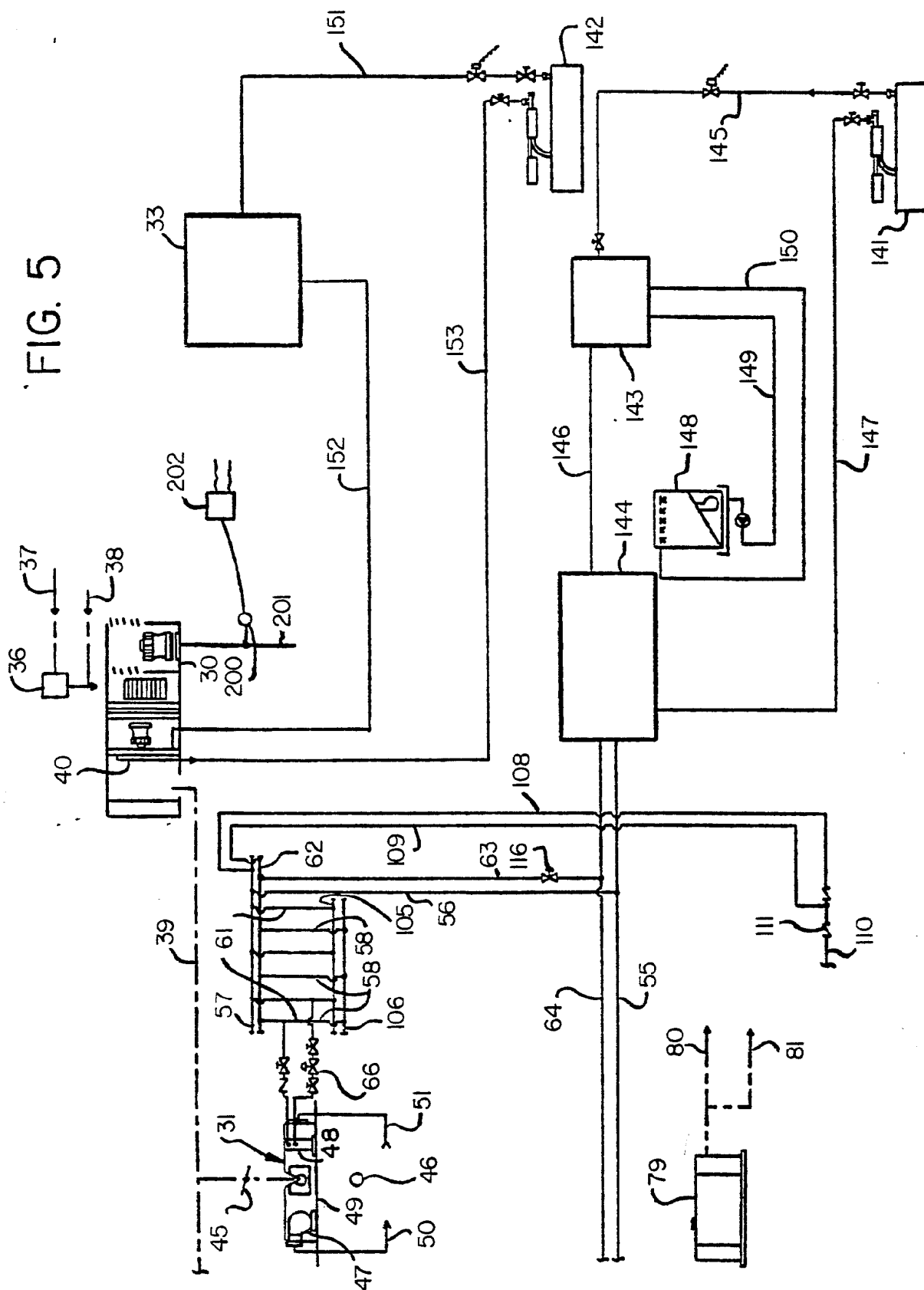


FIG. 5



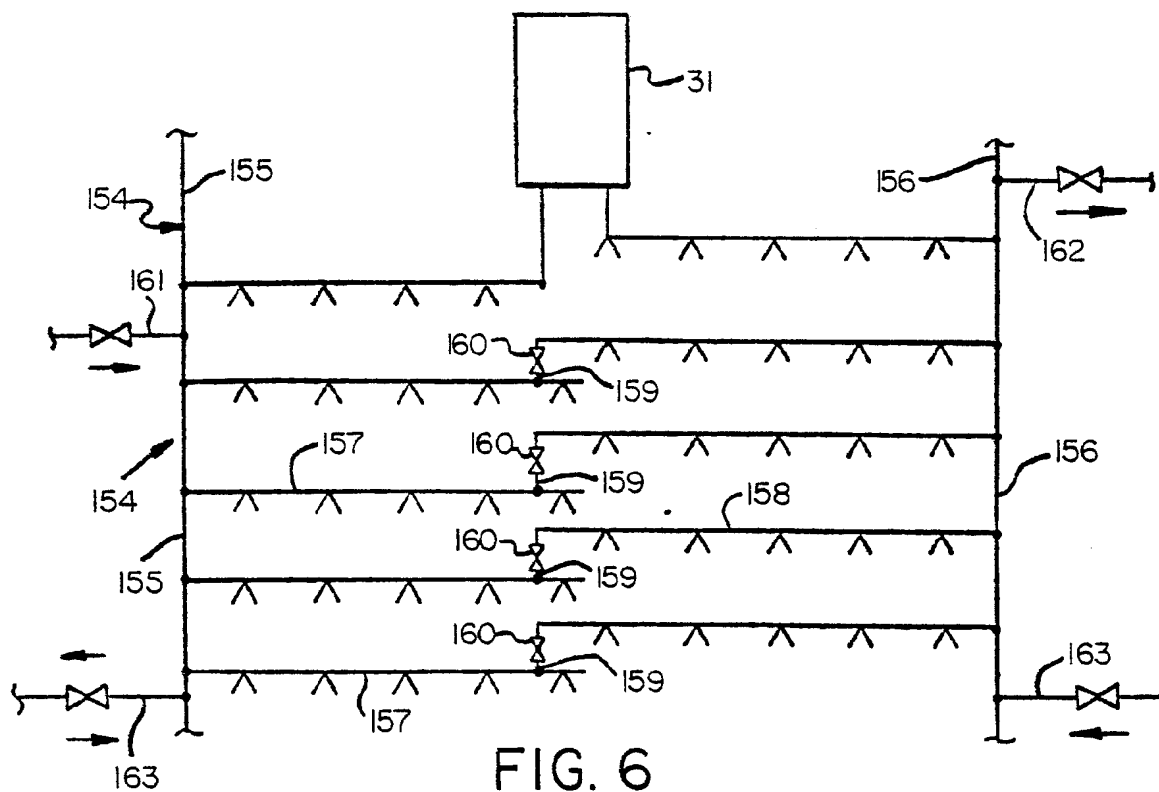


FIG. 6

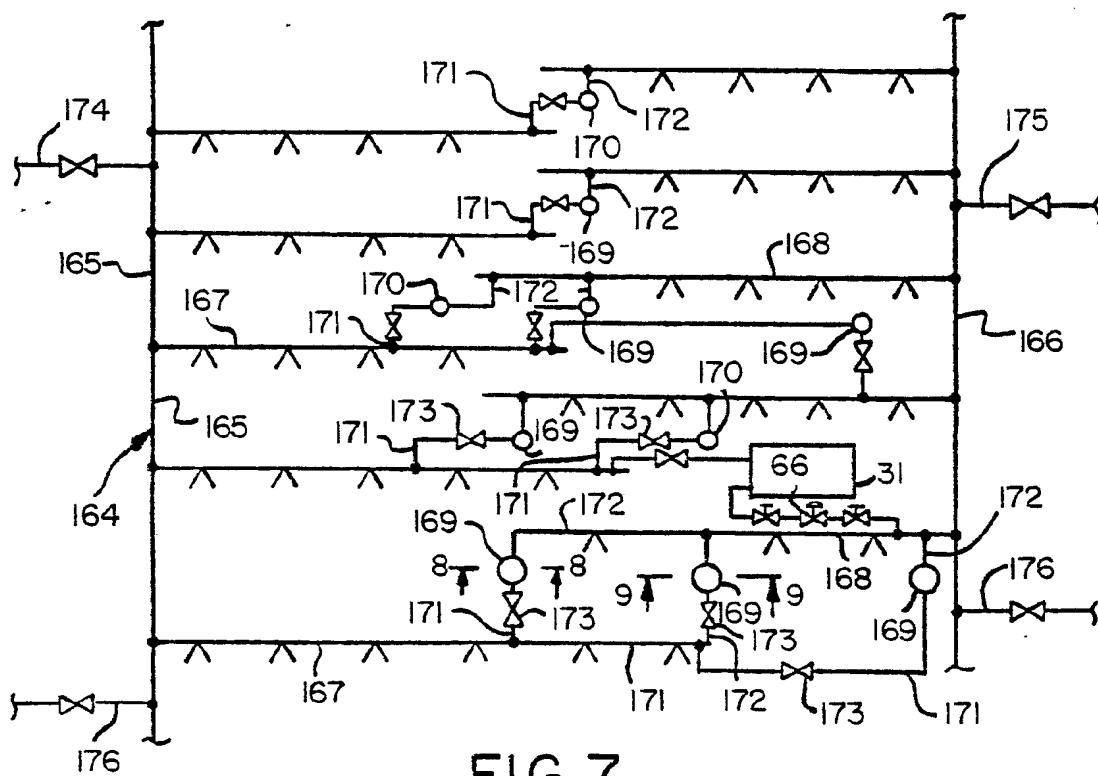


FIG. 7

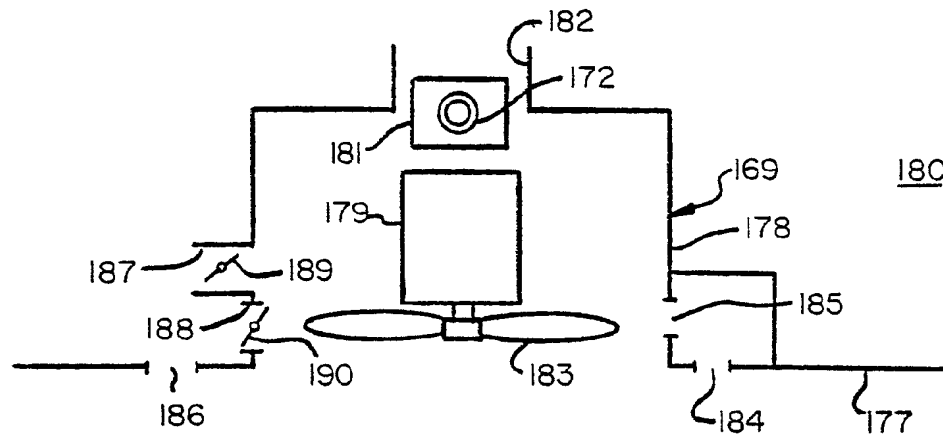


FIG. 8

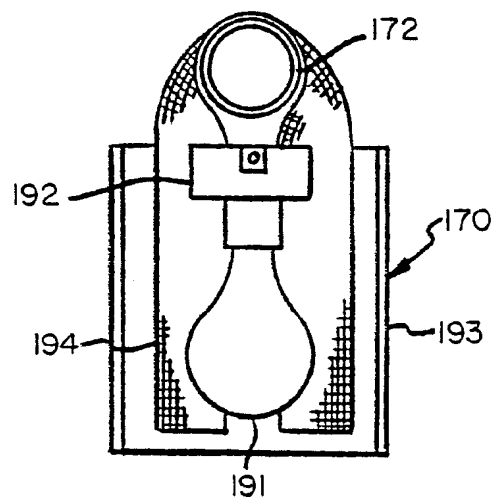


FIG. 9

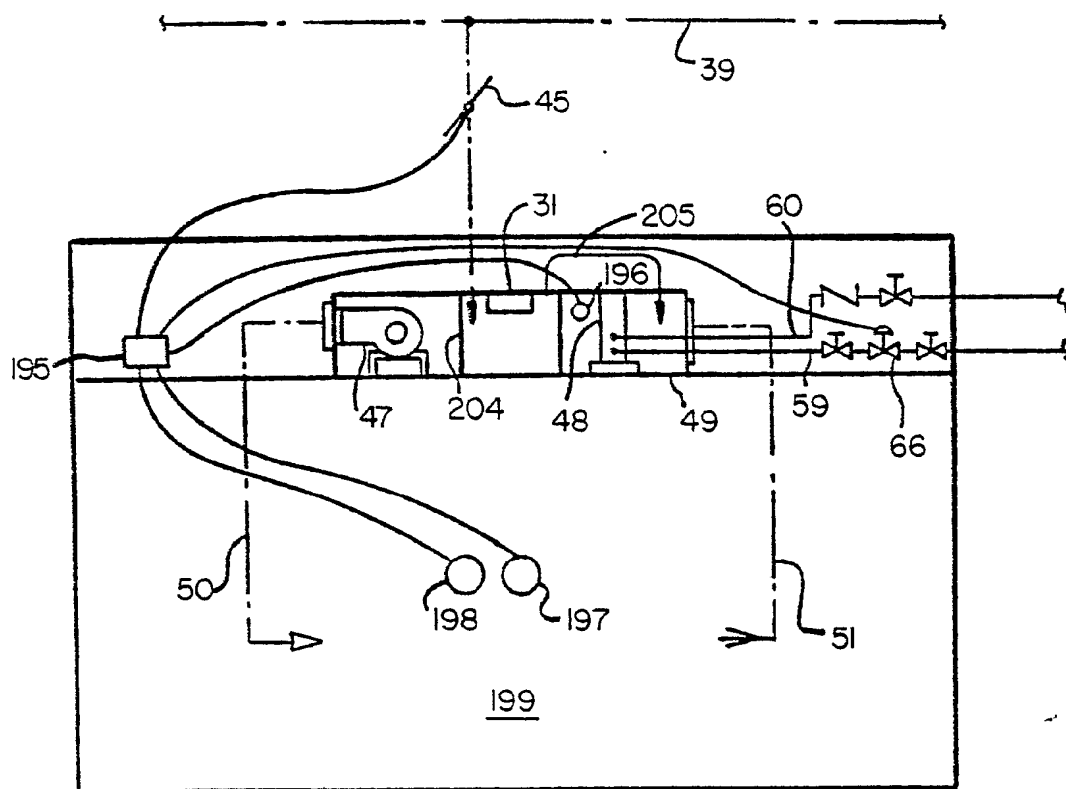


FIG. 11

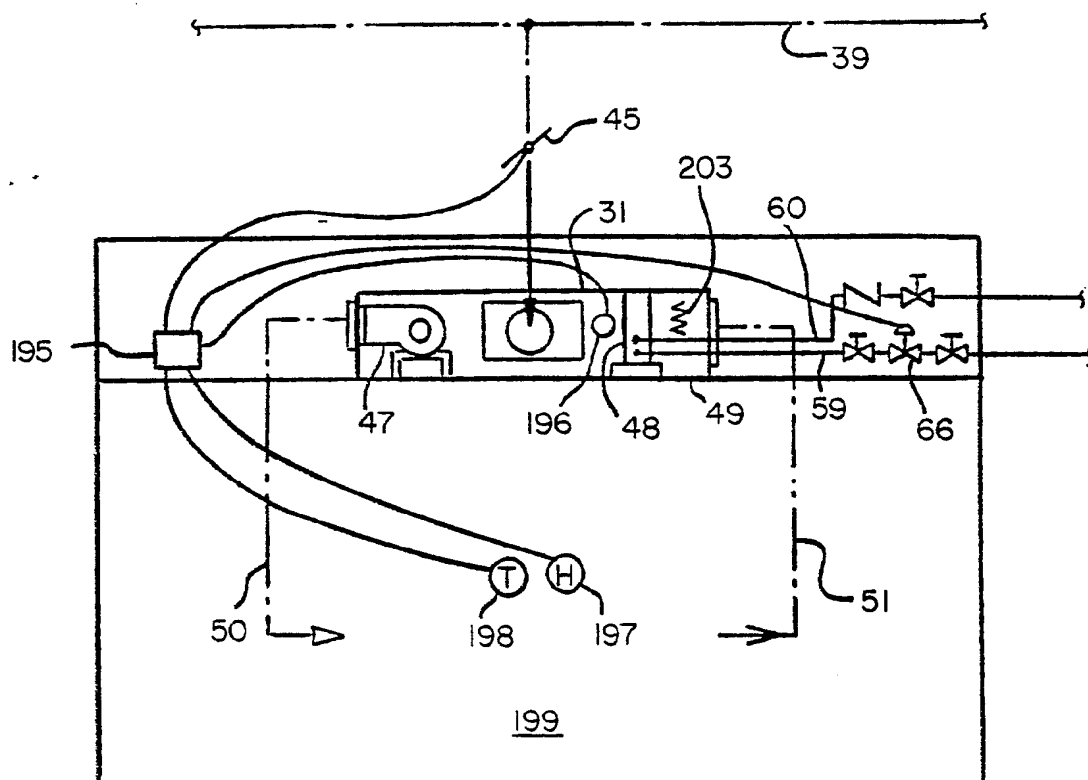
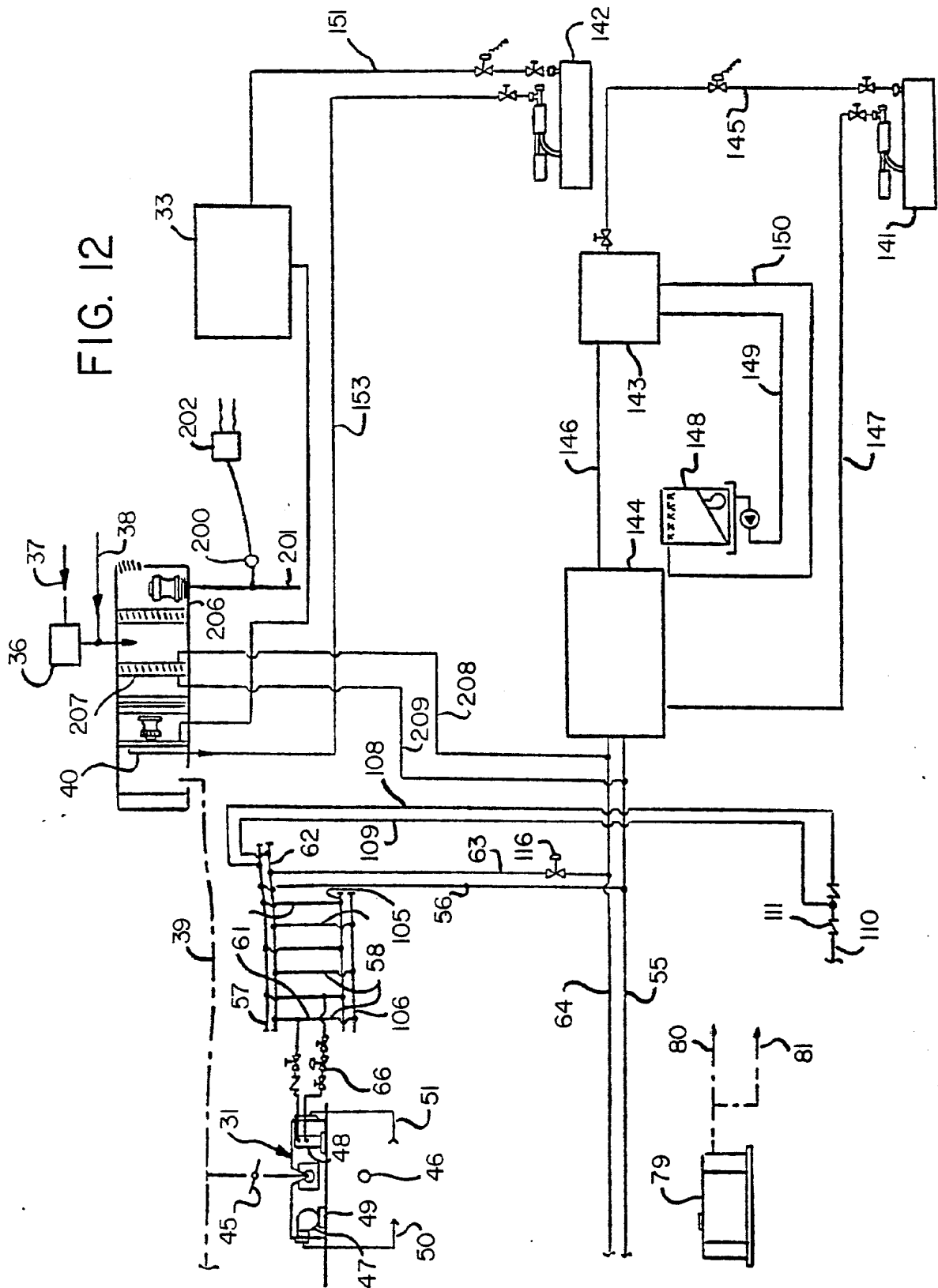


FIG. 10

FIG. 12



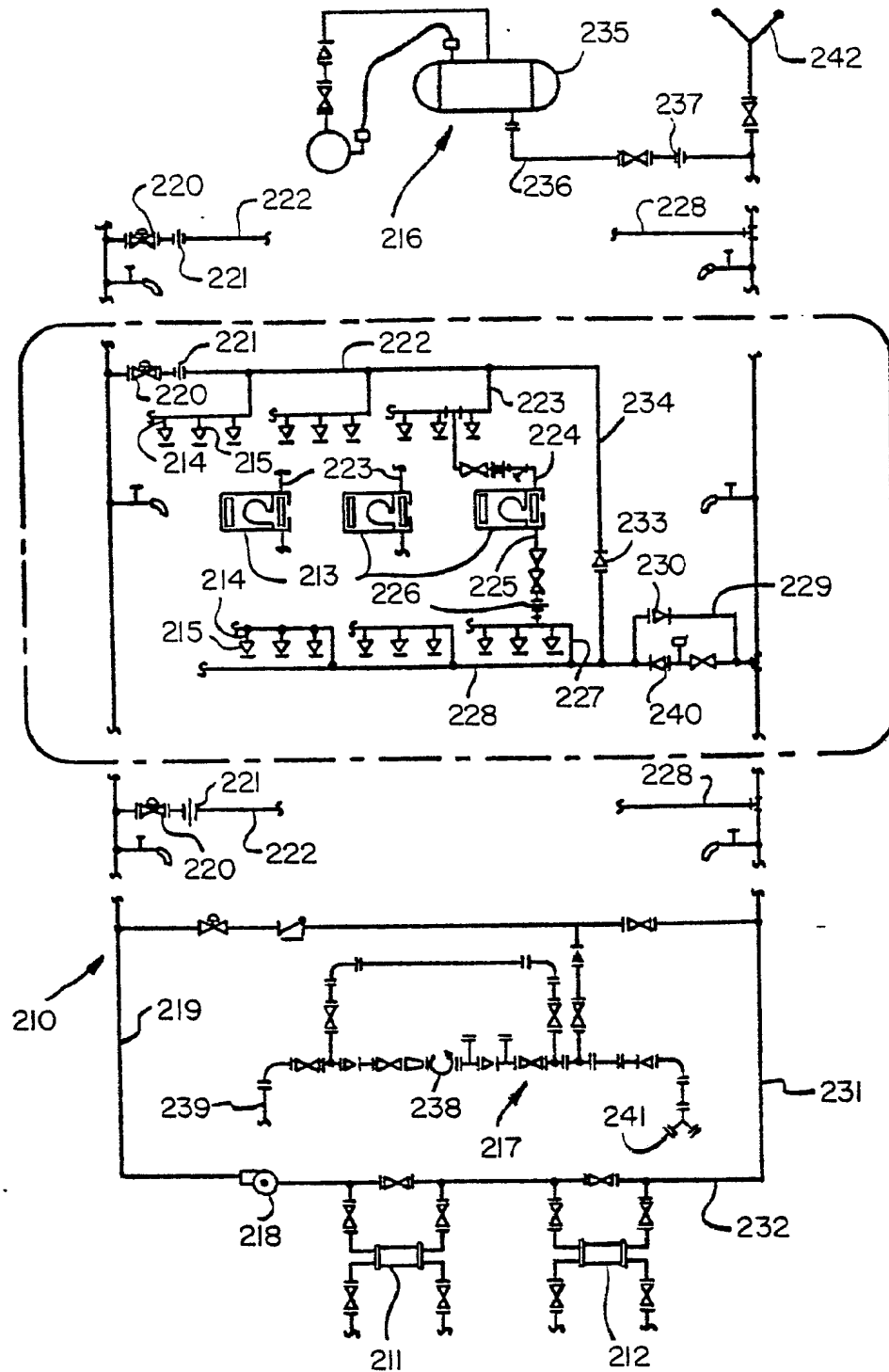


FIG. 13

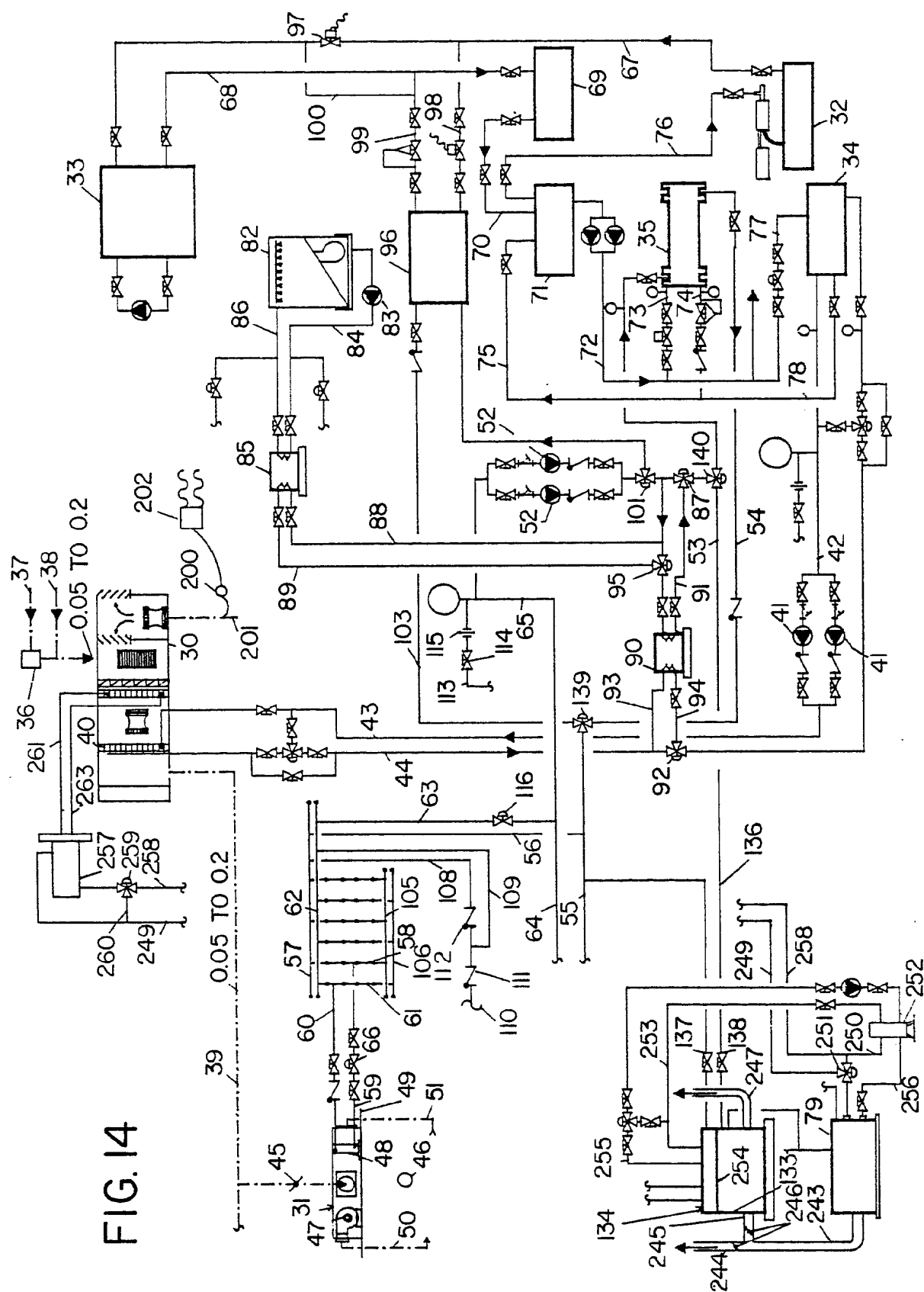


FIG. 16

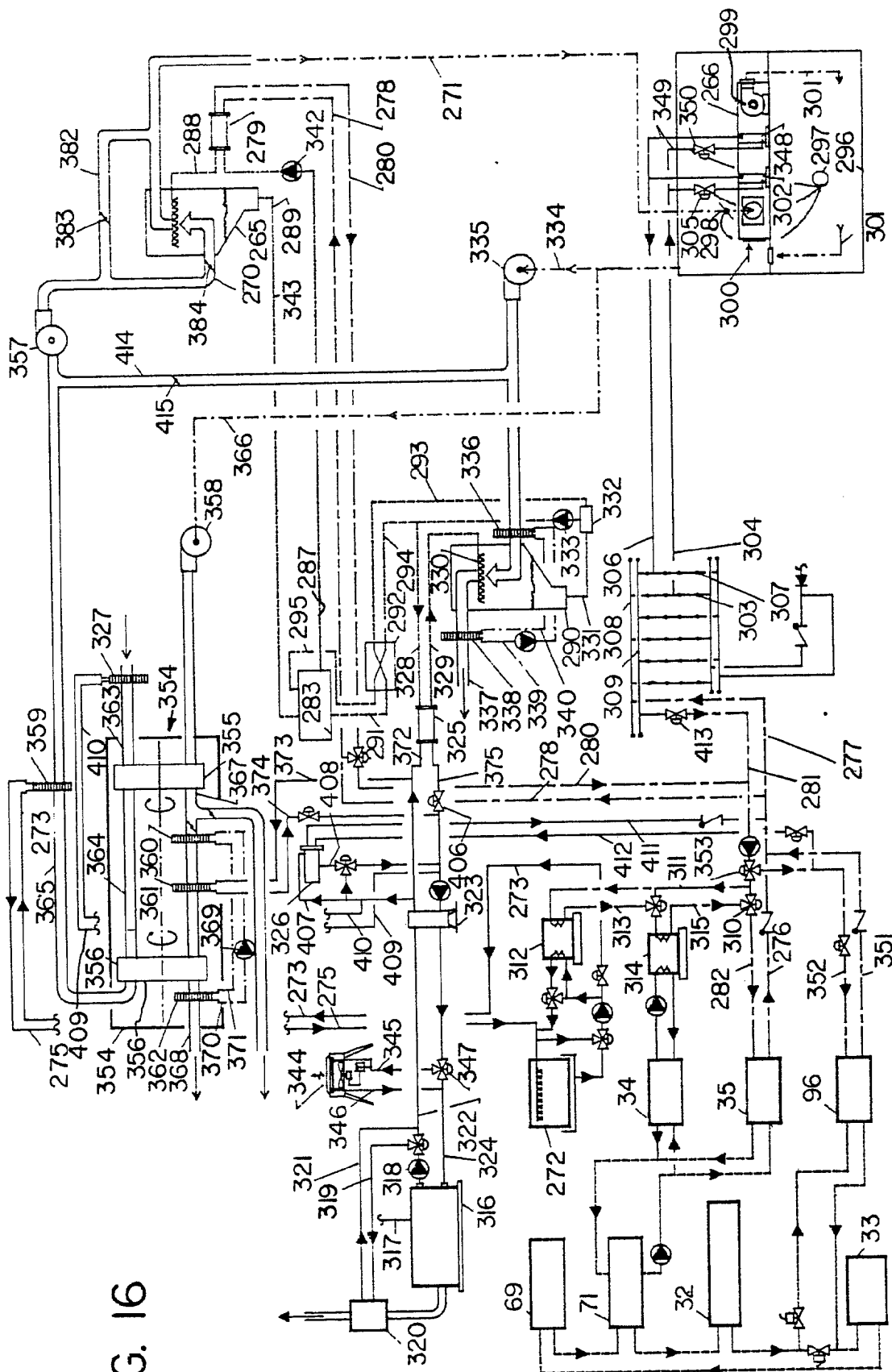


FIG. 17

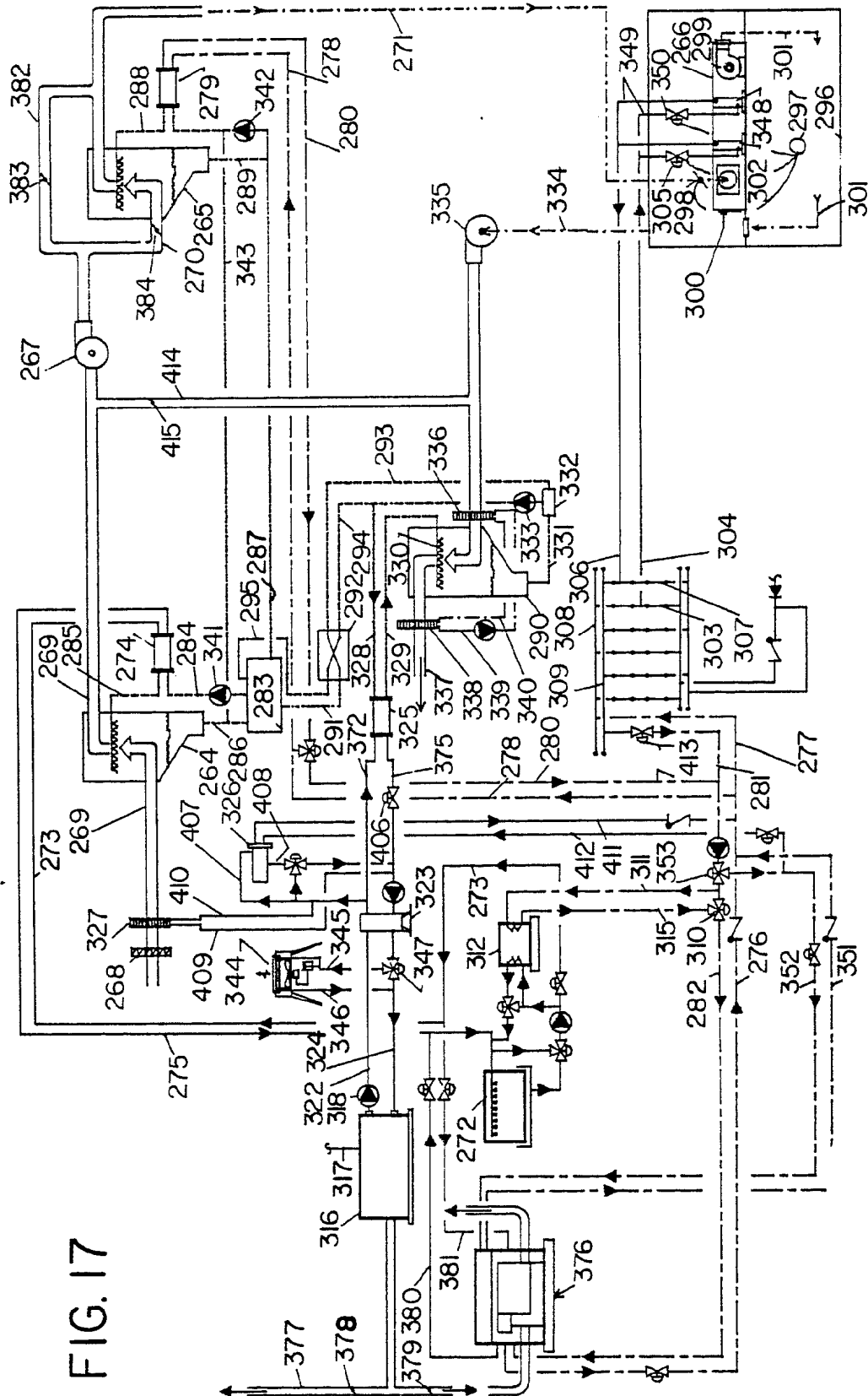
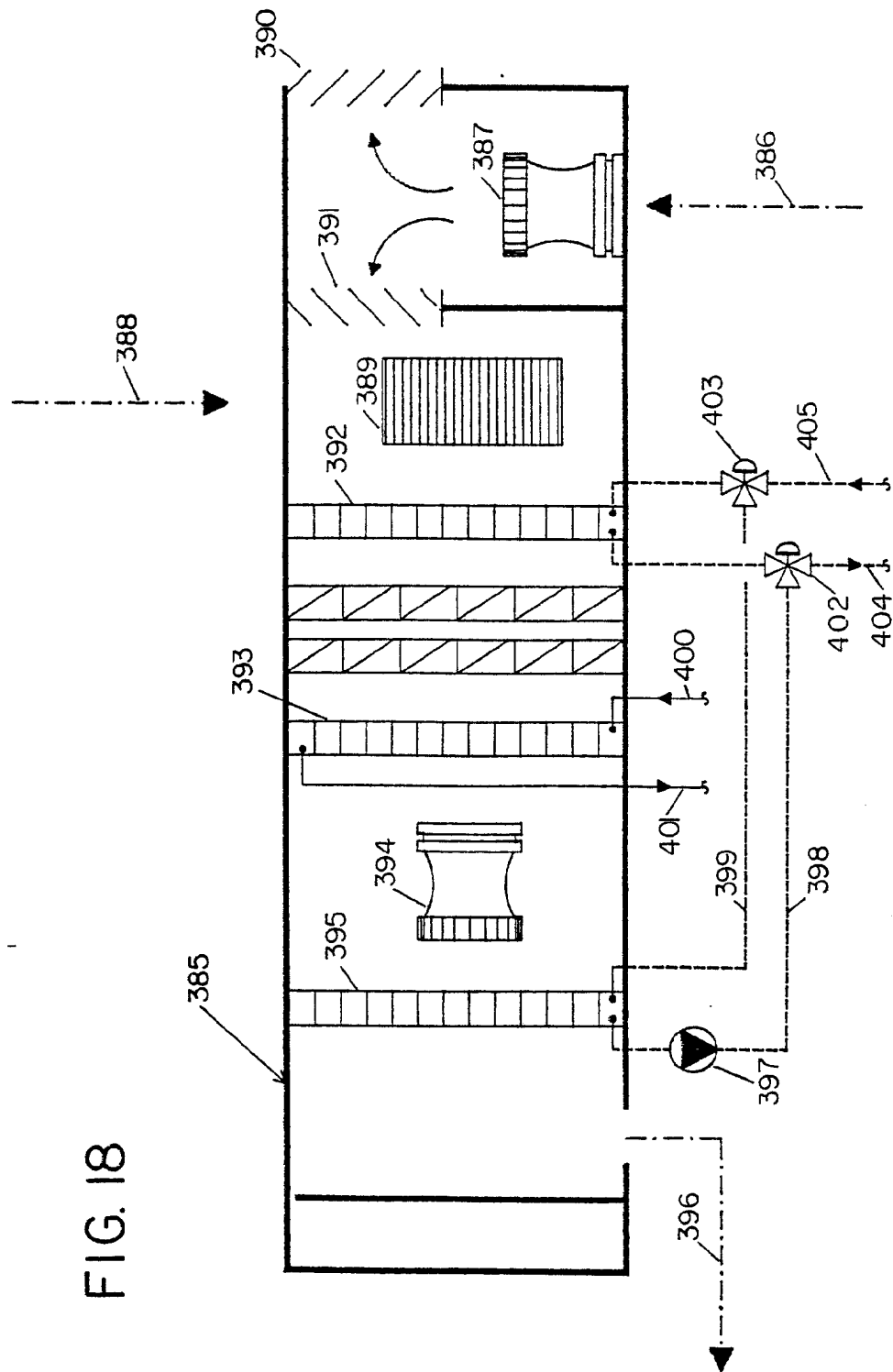
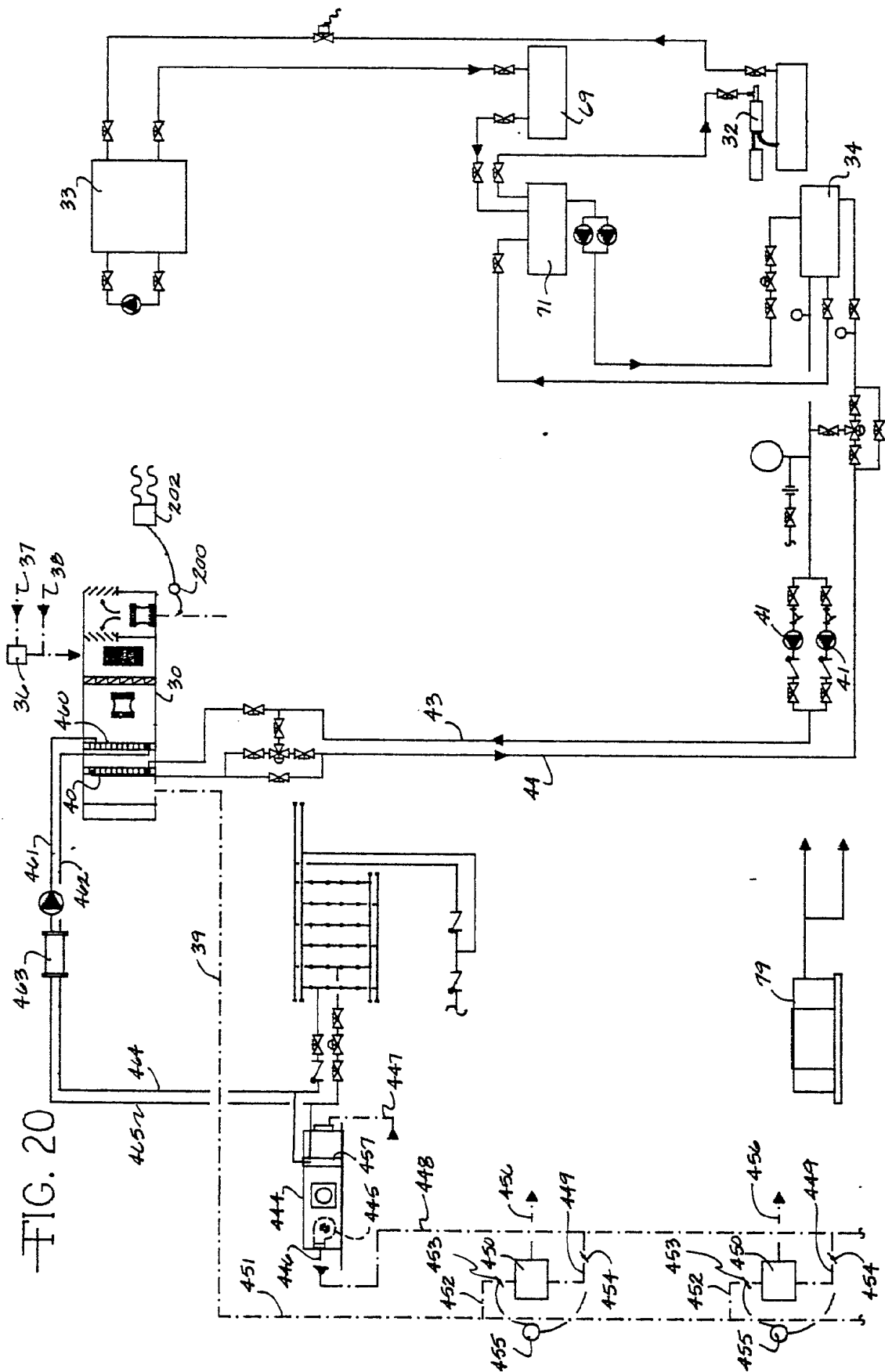


FIG. 18





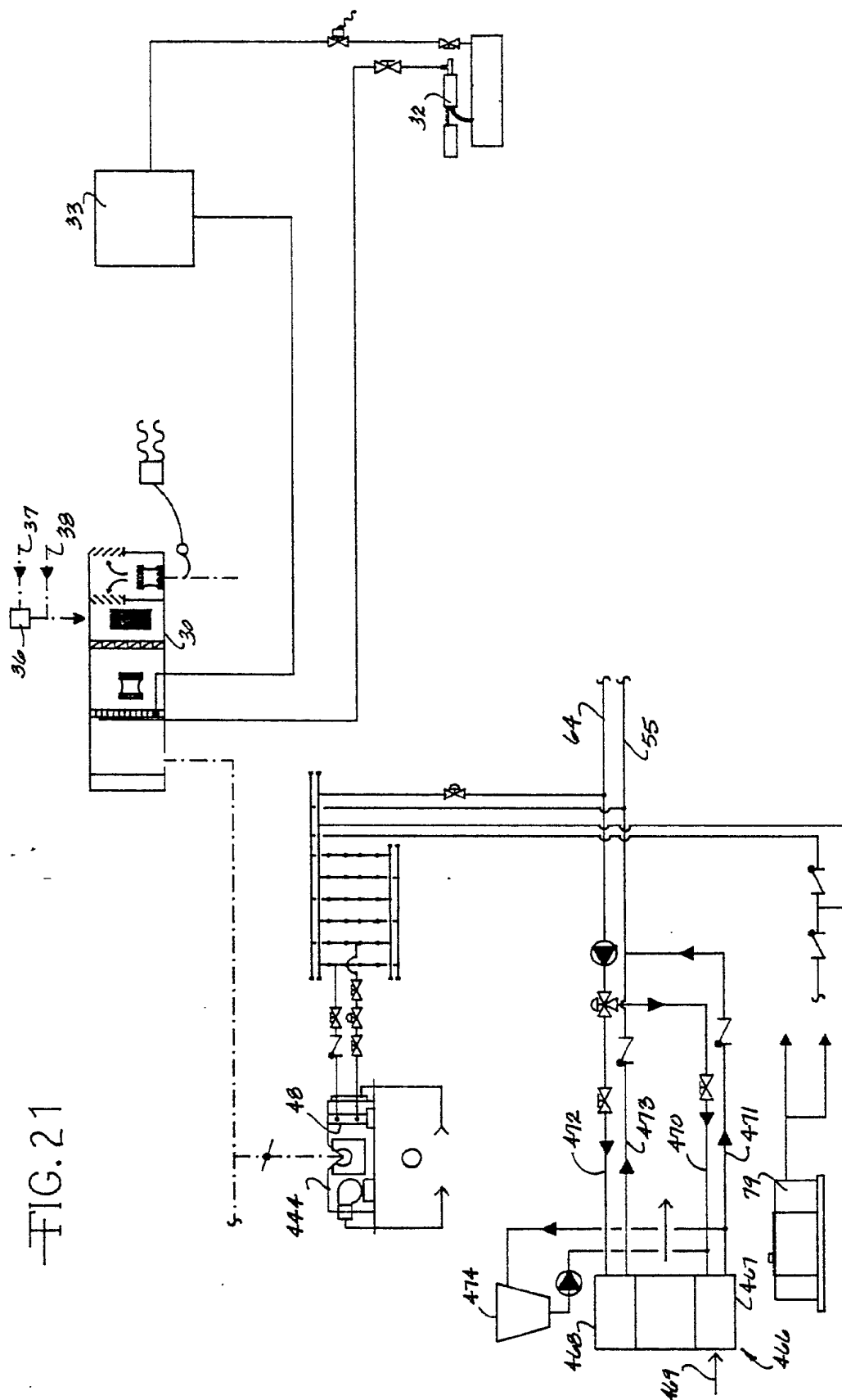


FIG. 21

FIG. 23

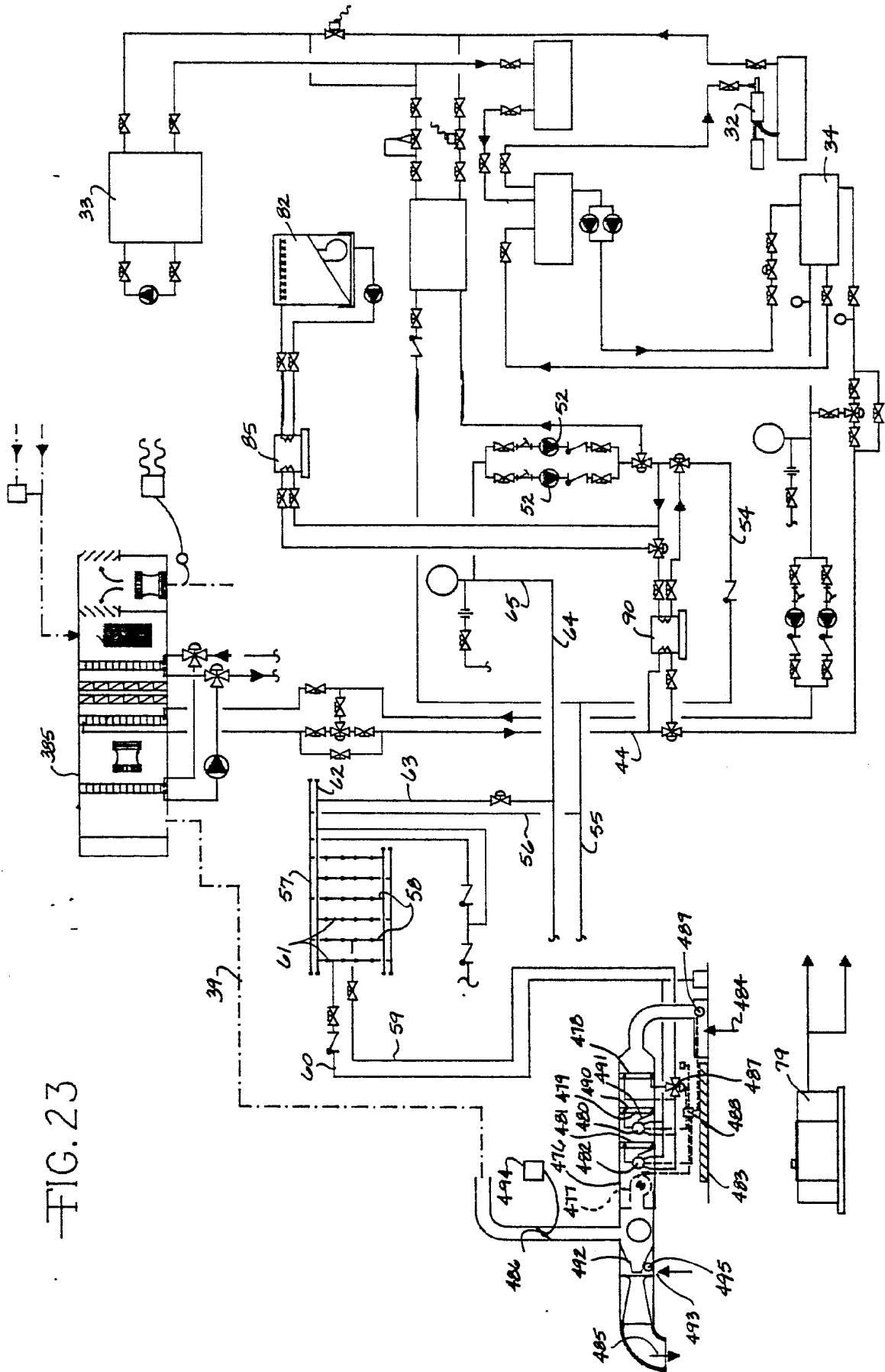
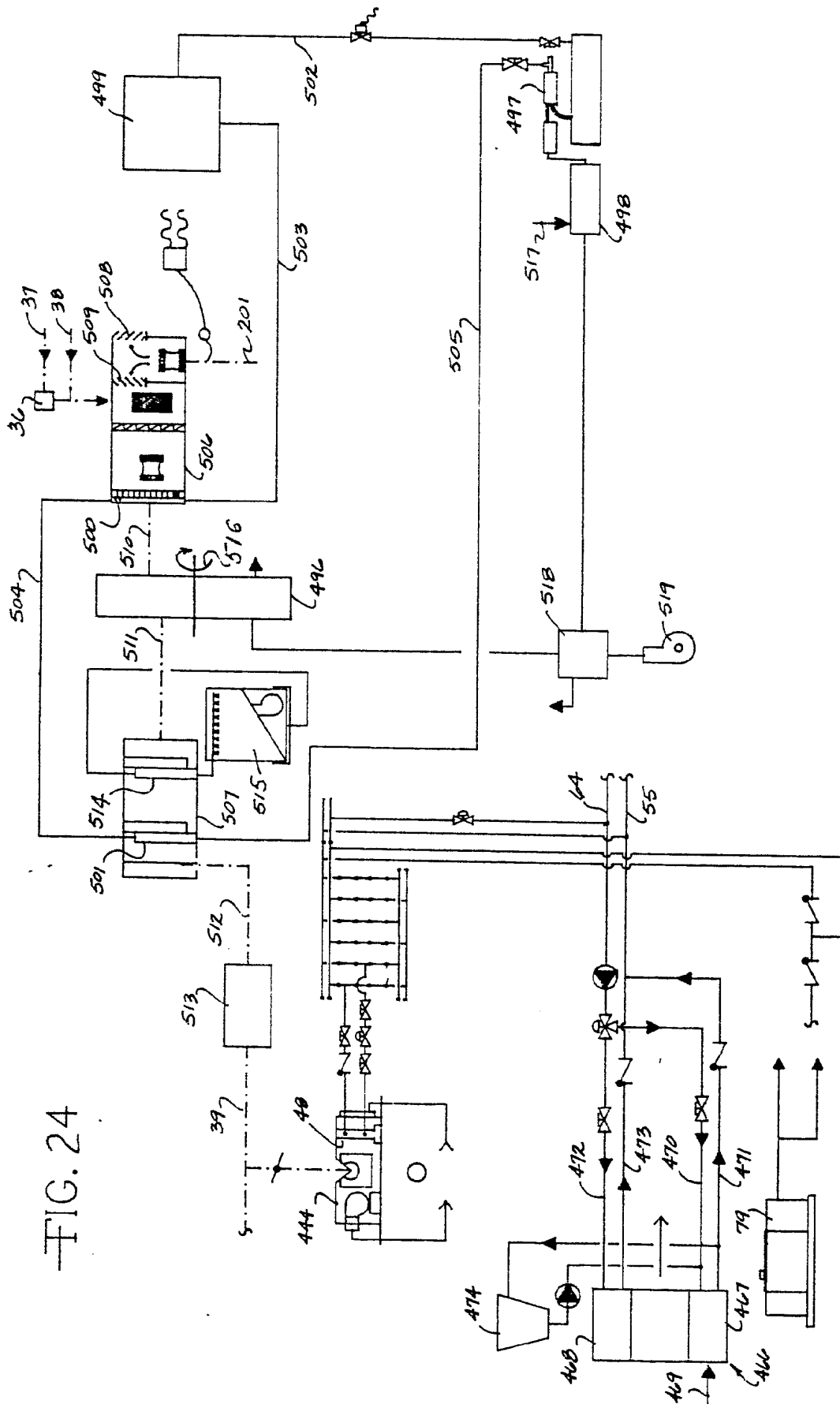


FIG. 24



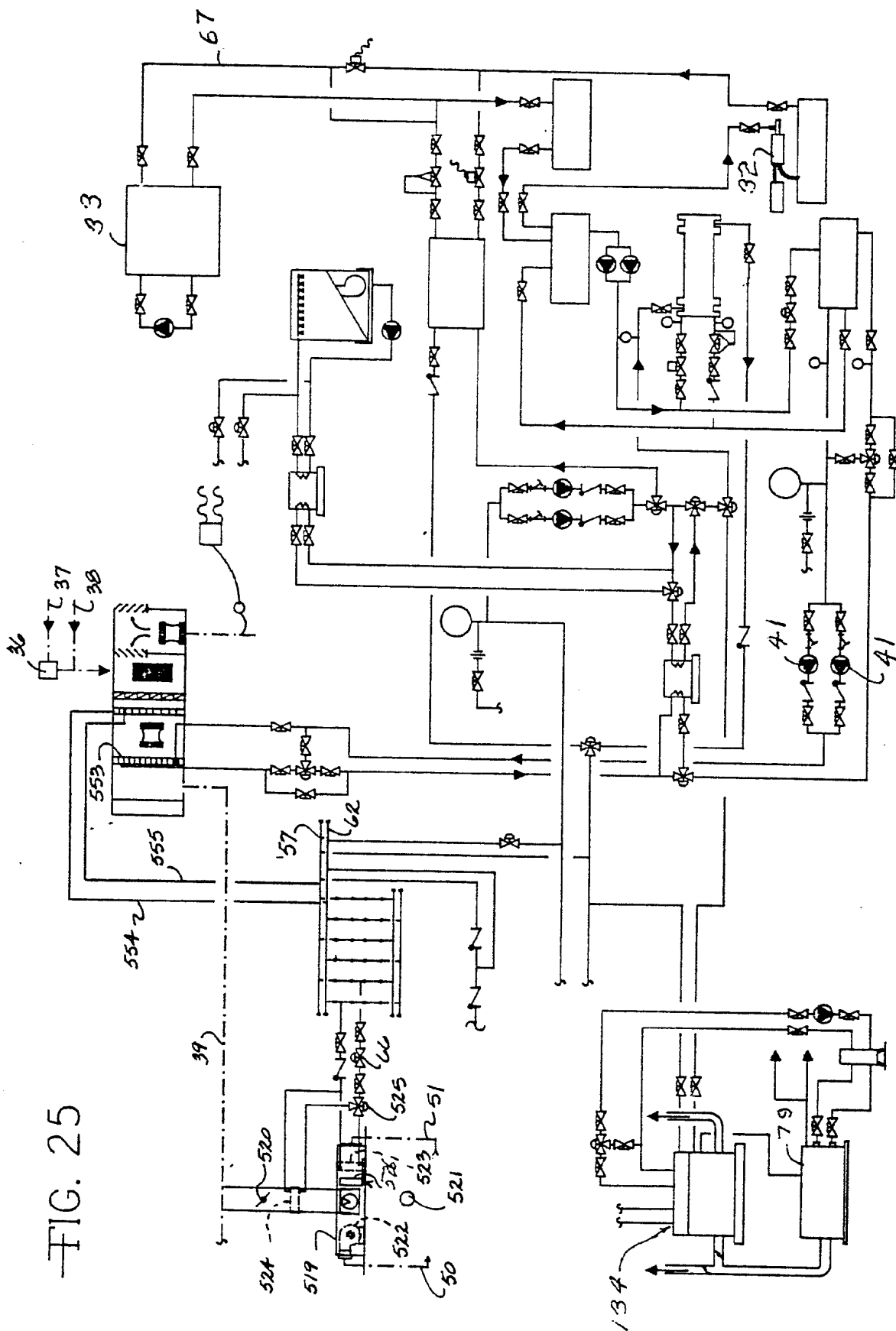


FIG. 25

FIG. 26

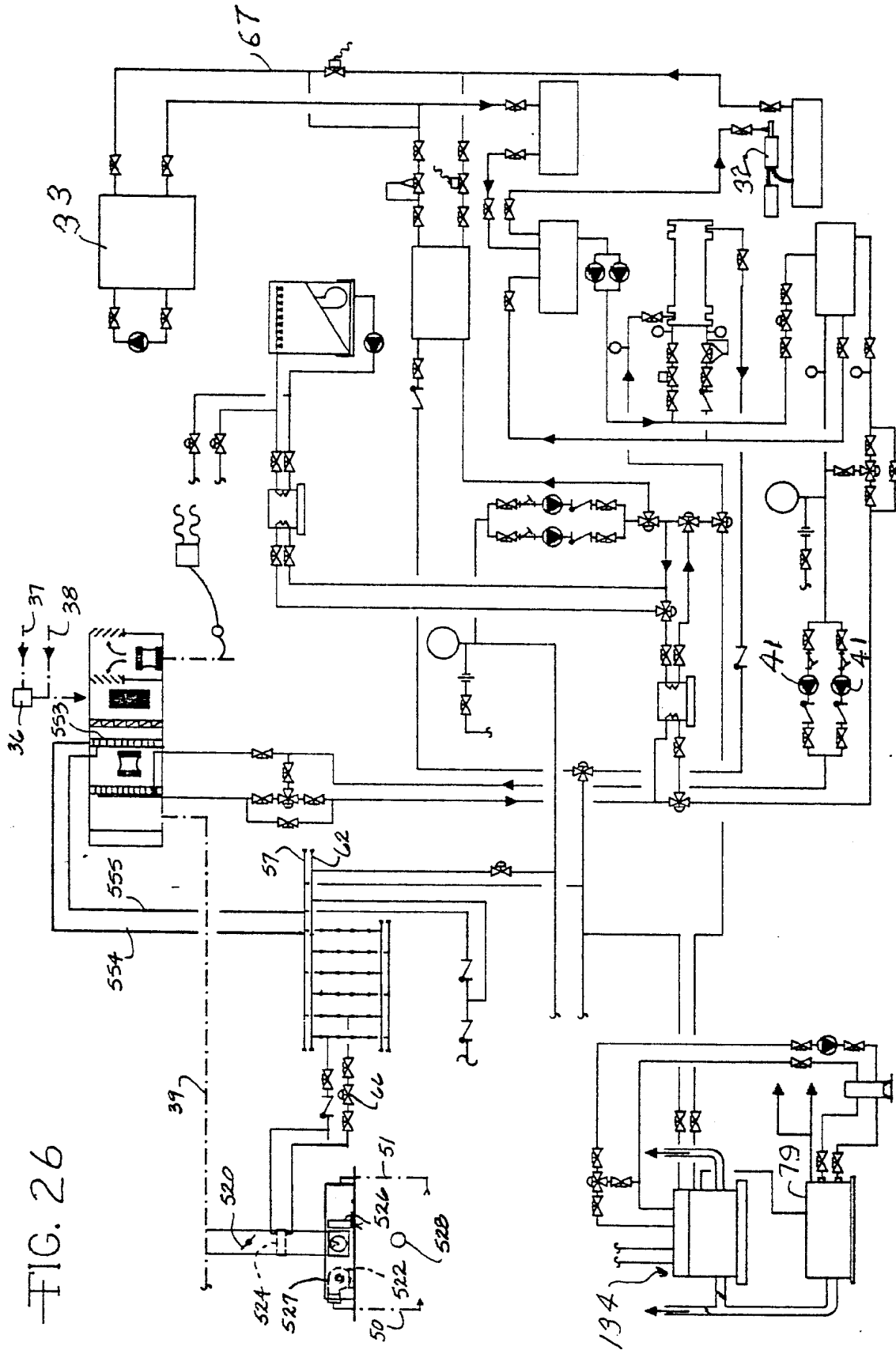


FIG. 27

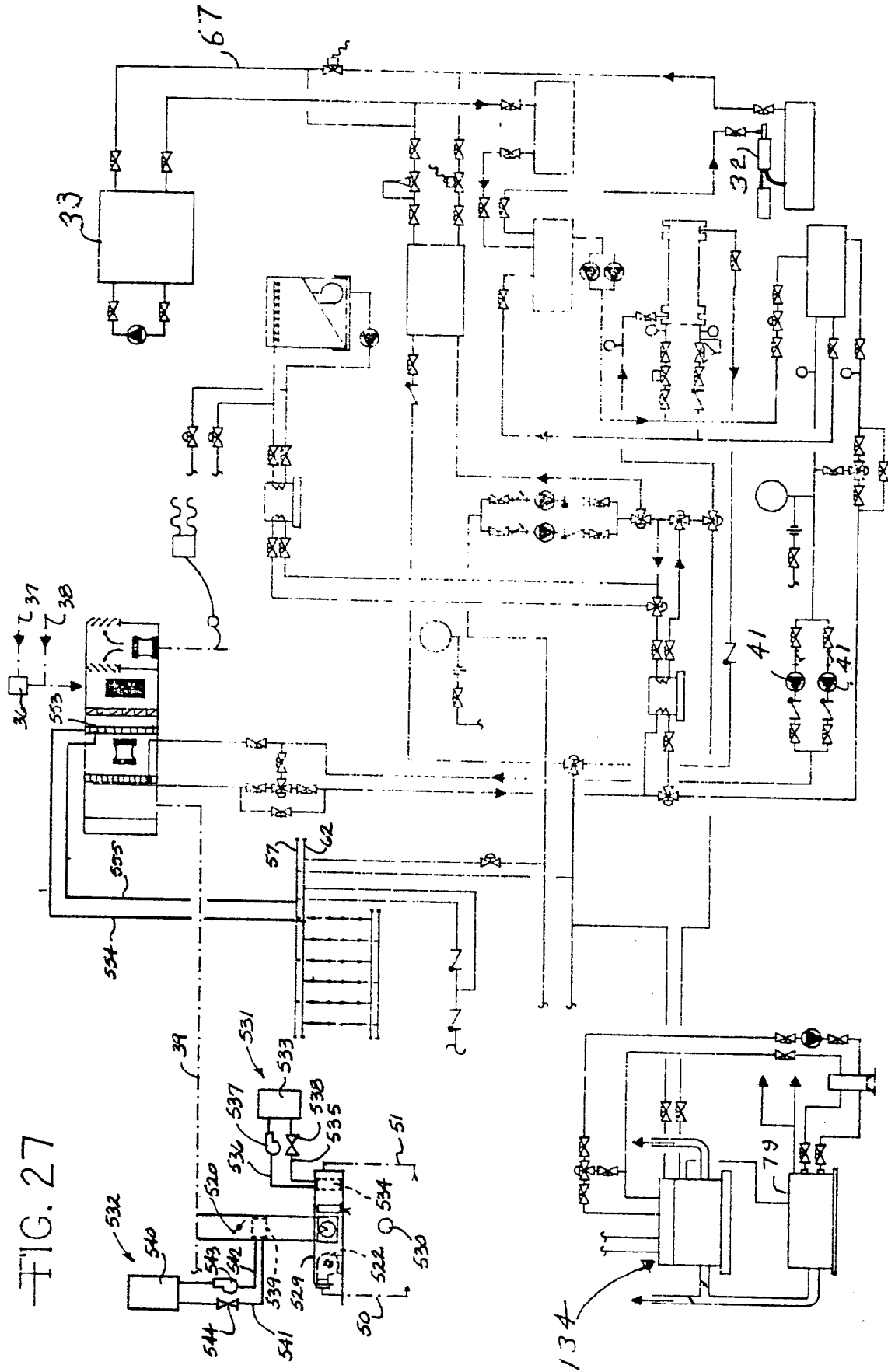
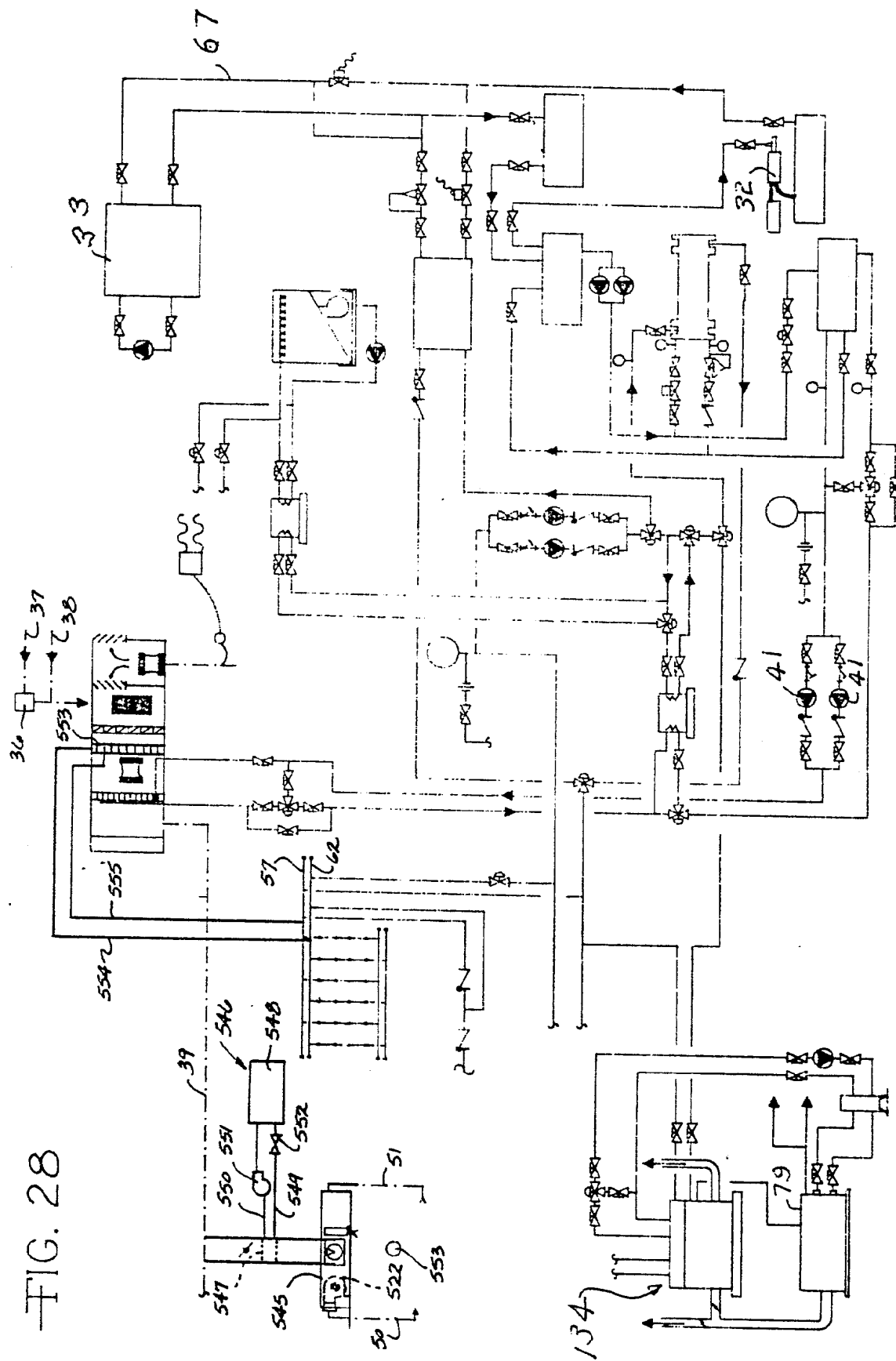


FIG. 28



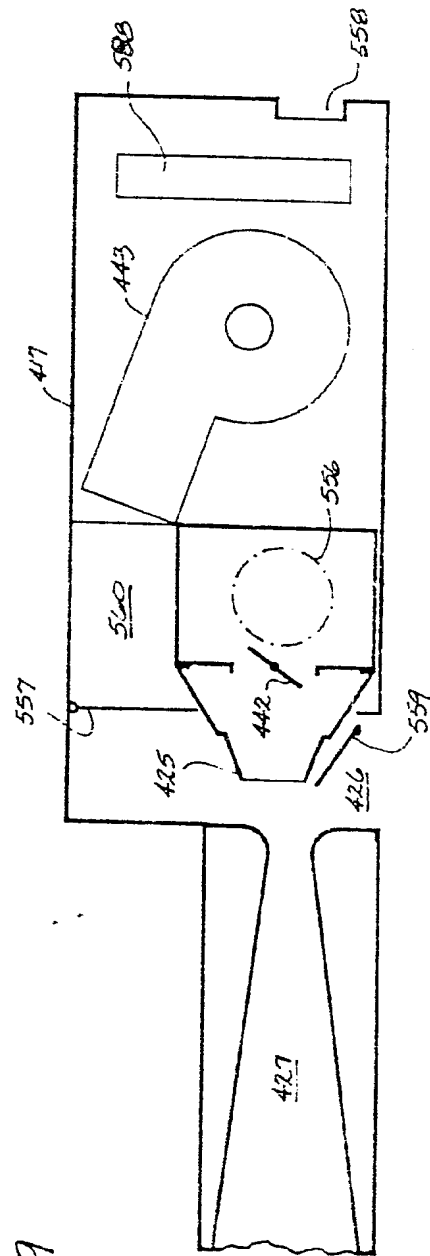


FIG. 29

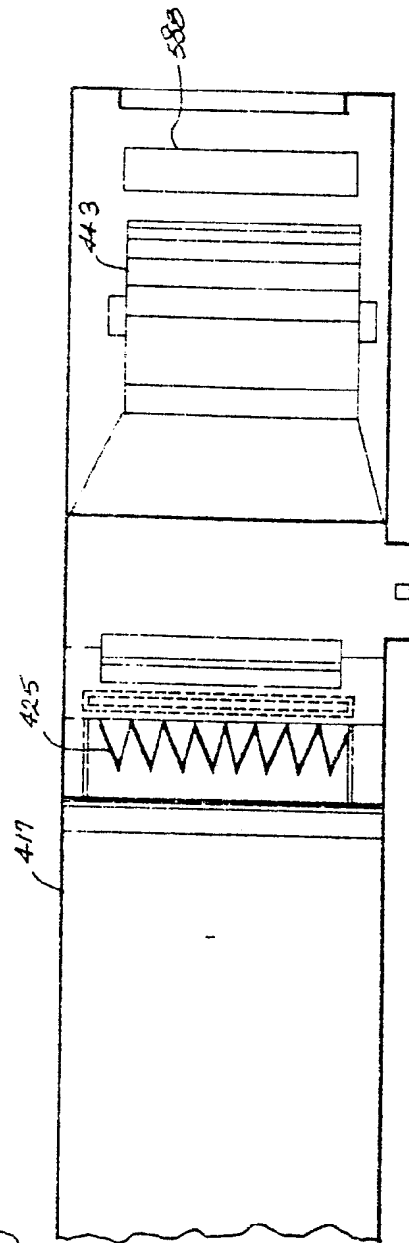


FIG. 30

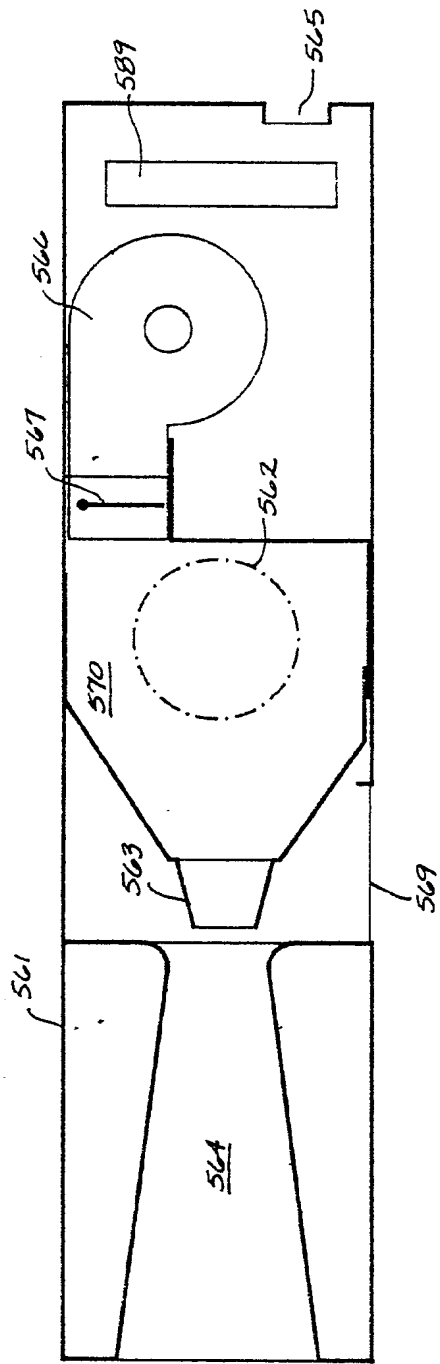


FIG. 31

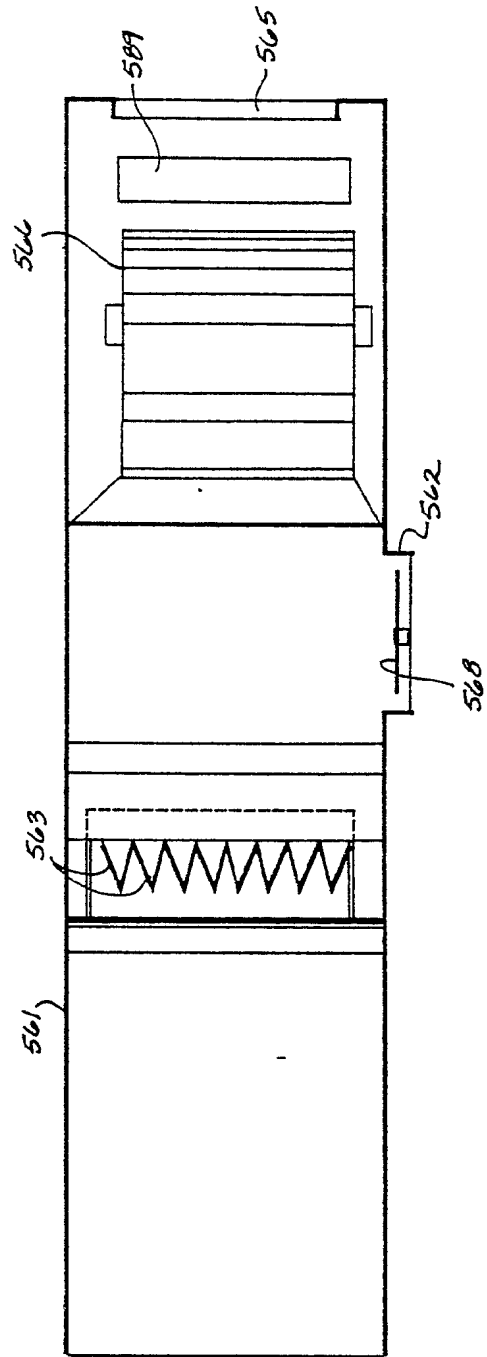
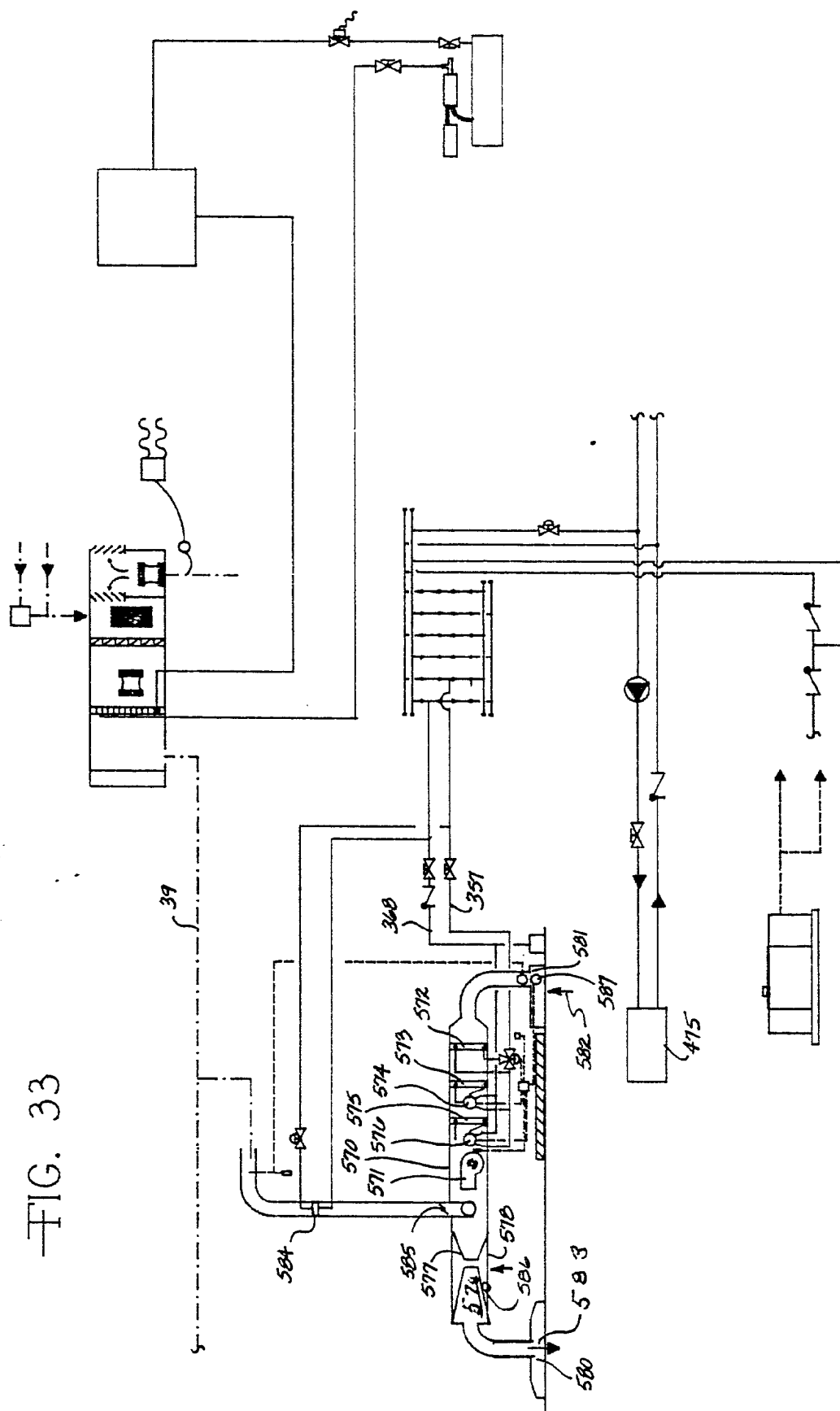


FIG. 32

FIG. 33



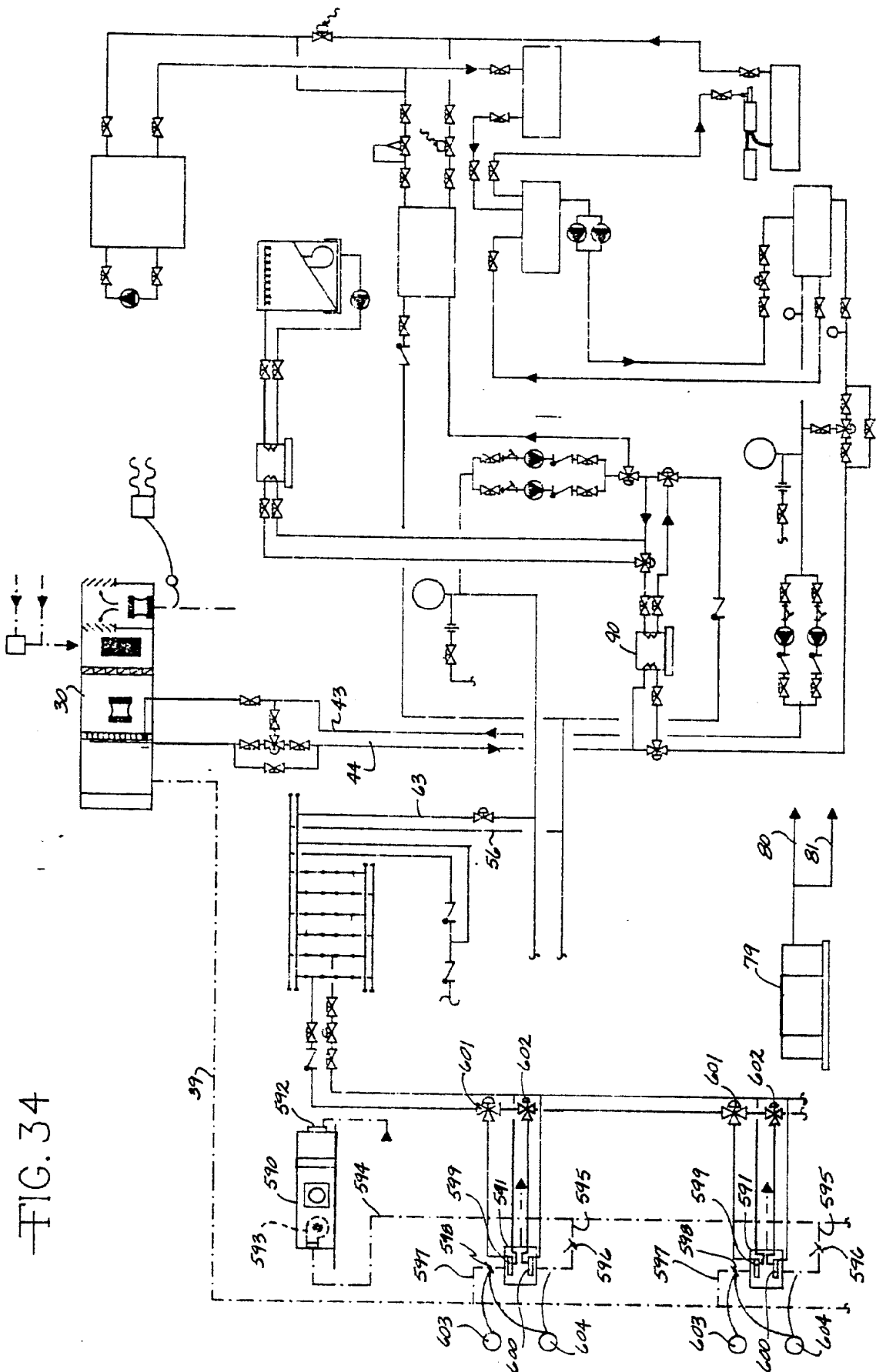


FIG. 34

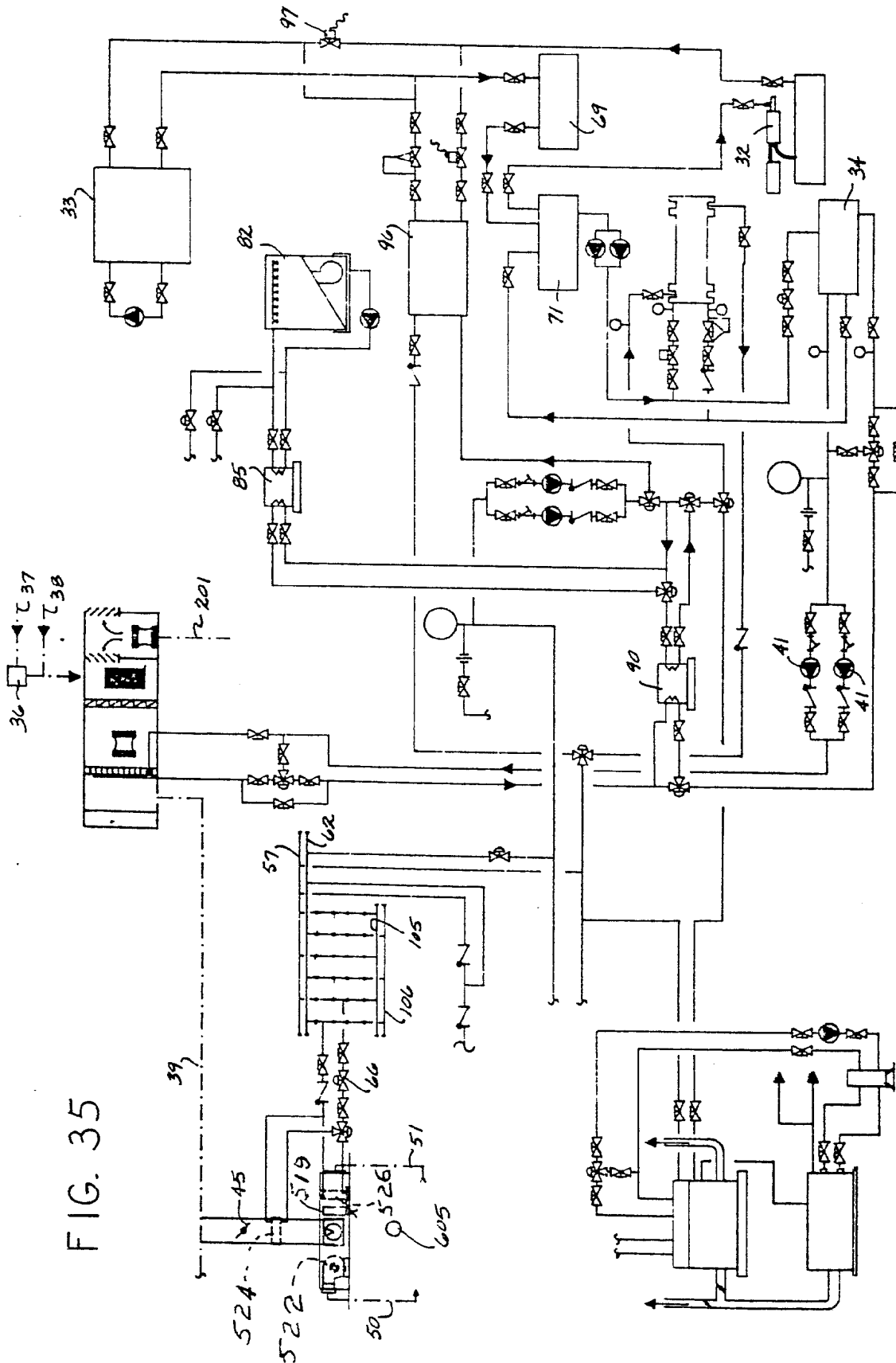
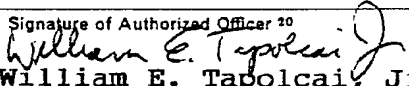


FIG. 35

INTERNATIONAL SEARCH REPORT

International Application No PCT/US87/00626

I. CLASSIFICATION OF SUBJECT MATTER (if several classification symbols apply, indicate all) ³		
According to International Patent Classification (IPC) or to both National Classification and IPC		
IPC (4): F24F 3/14; G05D 22/00		
U.S. CL. 62/176.4, 53; 165/19; 236/44A		
II. FIELDS SEARCHED		
Minimum Documentation Searched ⁴		
Classification System	Classification Symbols	
U.S.	62/92,93,176.4; 165/19,20; 236/44A	
Documentation Searched other than Minimum Documentation to the Extent that such Documents are Included in the Fields Searched ⁵		
III. DOCUMENTS CONSIDERED TO BE RELEVANT ¹⁴		
Category ⁶	Citation of Document, ¹⁶ with indication, where appropriate, of the relevant passages ¹⁷	Relevant to Claim No. ¹⁸
Y	US, A, 2,200,243 (NEWTON ET AL) 14 May 1940 See page 1, right column, line 45, to page 2, Left column, line 11.	1-5,14,23
Y	US, A, 2,262,954 (MATTERN ET AL) 18 November 1940 See page 1, right column, lines 35-52	1-5,14,15, 23
Y	US, A, 2,284,914 (MILLER) 02 June 1942 See page 1, right column, lines 22-52	1-5,14-24
Y	US, A, 2,290,465 (CRAWFORD) 21 July 1942 See page 1, right column, lines 4-42	1-5,7,14,15, 19,23
Y	US, A, 2,292,486 (STEINFELD) 11 August 1942 See page 1, right column, lines 35-48	7,8
Y	US, A, 3,247,679 (MECKLER) 26 April 1966 See column 2, lines 18-48	1-5,14-23
Y	US, A, 3,277,954 (MECKLER) 11 October 1966 See Column 4, lines 8-53	1-5,7,14,15, 19, 23
Y	US, A, 3,401,530 (MECKLER) 17 September 1968 See the Entire document	1-5,7,14,15, 23
<div style="display: flex; justify-content: space-between;"> <div style="width: 45%;"> <p>¹⁵ Special categories of cited documents:</p> <p>"A" document defining the general state of the art which is not considered to be of particular relevance</p> <p>"E" earlier document but published on or after the international filing date</p> <p>"L" document which may throw doubts on priority claim(s) or which is cited to establish the publication date of another citation or other special reason (as specified)</p> <p>"O" document referring to an oral disclosure, use, exhibition or other means</p> <p>"P" document published prior to the international filing date but later than the priority date claimed</p> </div> <div style="width: 45%;"> <p>"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention</p> <p>"X" document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step</p> <p>"Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art.</p> <p>"&" document member of the same patent family</p> </div> </div>		
IV. CERTIFICATION		
Date of the Actual Completion of the International Search ²	Date of Mailing of this International Search Report ³	
17 JULY 1987	23 JUL 1987	
International Searching Authority ¹	Signature of Authorized Officer ²⁰	
ISA/US	 William E. Tapolcai, Jr.	

III. DOCUMENTS CONSIDERED TO BE RELEVANT (CONTINUED FROM THE SECOND SHEET)		
Category *	Citation of Document, ¹⁶ with indication, where appropriate, of the relevant passages ¹⁷	Relevant to Claim No ¹⁸
Y	US, A, 3,403,723 (MECKLER) 01 October 1968 See column 2, line 66 to column 3, line 11, and column 8, lines 15-32.	1-5,7,14,15, 23
Y	US, A, 4,164,125 (GRIFFITHS) 14 August 1979 See Column 3, line 21 to column 4, line 48	1-5,7,14,15, 23
Y	US, A, 4,222,244 (MECKLER) 16 September 1980 See column 3, line 20 to column 4, line 49.	1-5,7,14,15, 23

FURTHER INFORMATION CONTINUED FROM THE SECOND SHEET

V. ☐ OBSERVATIONS WHERE CERTAIN CLAIMS WERE FOUND UNSEARCHABLE ¹⁰

This international search report has not been established in respect of certain claims under Article 17(2) (a) for the following reasons:

1. ☐ Claim numbers because they relate to subject matter ¹² not required to be searched by this Authority, namely:

2. ☐ Claim numbers because they relate to parts of the international application that do not comply with the prescribed requirements to such an extent that no meaningful international search can be carried out ¹³, specifically:

VI. ☒ OBSERVATIONS WHERE UNITY OF INVENTION IS LACKING ¹¹

This International Searching Authority found multiple inventions in this international application as follows:

- I. Claims 1-26 drawn to a dehumidification system with humidity sensor and control; class 62 subclass 176.4
- II. Claim 27 drawn to a sensor and control for discharging dehumidified air; class 236 subclass 44A.
- III. Claims 28-44 drawn to an air mixing and control system; class 236 subclass 13.

1. ☐ As all required additional search fees were timely paid by the applicant, this international search report covers all searchable claims of the international application.

2. ☐ As only some of the required additional search fees were timely paid by the applicant, this international search report covers only those claims of the international application for which fees were paid, specifically claims:

3. ☒ No required additional search fees were timely paid by the applicant. Consequently, this international search report is restricted to the invention first mentioned in the claims; it is covered by claim numbers:

1-26

4. ☐ As all searchable claims could be searched without effort justifying an additional fee, the International Searching Authority did not invite payment of any additional fee.

Remark on Protest

☐ The additional search fees were accompanied by applicant's protest.

☐ No protest accompanied the payment of additional search fees.